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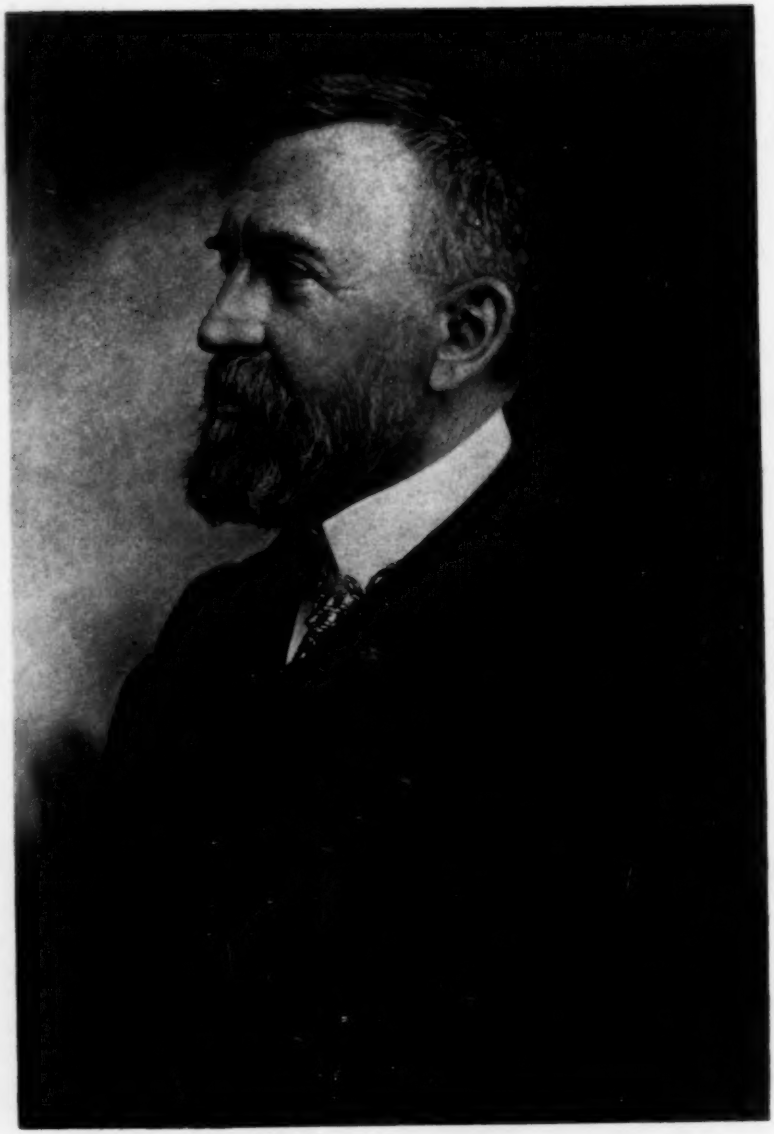

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CONTENTS

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HISTORY OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Preliminary Report of the Committee on Society History

Chapter IX

GROWING INFLUENCE OF THE SOCIETY

217 In looking back over the development of The American Society of Mechanical Engineers from the first small beginning in the third floor room at the corner of Fulton and William streets to a strong and powerful organization, owning its own house in the center of New York City, and forming one of the central features at an international congress at the World's Fair, we realize the extent to which the Society is indebted to the efforts of a number of devoted men.

218 It has been said that it is only within recent years the engineer has taken his place as a business man as well as the creator of structures of stone, wood, iron, and steel, and as the manufacturer of power; but it is none the less true that in the years from 1881 to 1893, administrative qualities of a high order were shown in the development of the Society.

219 In its modest beginning, the headquarters of the Society was the office of the Secretary, a room of sufficient size for the small amount of business to be transacted, and for the meetings of the Council, this office being at the corner of Broadway and Park Place, in a building still standing (1909), although materially altered.

220 Upon the election of Prof. F. R. Hutton as Secretary in 1883

Under the direction of the Council the Committee on Society History has arranged to present the results of its investigations to the members of the Society.
The Preliminary Report will appear in The Journal of the Society from month to month, and thus enable the matter to be open to comment during its completion. It is especially desired that any member who may be in the possession of facts or information bearing upon the various points as they are thus made public will communicate with the committee, in order that the final and completed report may have the advantage of the collaboration of the membership at large.

the office was removed to No. 15 Cortlandt St., near Broadway; and in 1885 it was again moved to No. 280 Broadway, in the building originally constructed by the late A. T. Stewart as his downtown store, and later converted into a business building.

221 It was in this latter location that the first plans for the formation of a library were consummated; and the request for contributions of books, trade catalogues and periodicals resulted in the filling of a bookcase, which, with the successors soon demanded, was the origin of the fine collection of books now forming a portion of the great engineering library which crowns the building in Thirty-ninth Street.

222 During those early days the question of the finances of the Society often formed the subject of earnest thought and discussion, and although the budget which then had to be met would look small in comparison with that of the present, it was more than once a weighty question for officers, committee and members. A glance over the early volumes of the Transactions will show the extent of the burden which the young organization had assumed.

223 With the growth in membership the number and extent of papers increased, and in a few years the size of the yearly volume fully equalled that issued by older organizations.

224 The names there met most frequently include some no longer living, and the work of Holley, Thurston, Wood, Hoadley, Worthington, Holloway, Babcock, and others remains to show the extent to which those devoted men gave time, effort and means to the organization.

225 It must be remembered that the period from the foundation of the Society to the World's Fair of 1893 was one in which many radical transformations in the work of the mechanical engineer were occurring. The general introduction of steel in the place of iron both for machine and building construction took place within that decade. The high-speed engine was developed to meet the demands of the electrical industry. The electrical industry itself was created in the course of those eventful years. Visitors to the Electrical Exhibition held in Philadelphia in 1884 will remember the Edison "Jumbo" dynamo as one of the wonders of the show, while those who saw it again, perched in the Electrical Building of the World's Fair at Chicago looked at it as a relic of a past age, as one gazes at the fragments of Stephenson's "Rocket" in the South Kensington Museum.

226 Practically the entire development of automatic machine-

tools occurred between the same limits of time, and indeed the transformation in machine tools generally which preceded the introduction of modern rapid cutting steels was the result of the efforts of men who were working also in and for the Society. Not alone in the electrical industry but in almost every other department of engineering work the enormous progress appearing between the mechanical exhibits at Philadelphia in 1876 and Chicago in 1893 was due in large degree to the genius and hard work of the members of the Society.

227 In the development of engineering education also the Society bore an important part and with that phase of engineering work it was identified from the start. From the organization meeting in the hall of the Stevens Institute in Hoboken, aided by the efforts of Professor Thurston and of Dr. Henry Morton, the Society has been an effective element in the progress of engineering education.

228 The ranks of the Junior membership were continually supplied from the graduates of the technical schools, while professors of engineering were included among its active contributors to the Transactions and on the floor at its conventions. To this firm bond of union between the teachers and the manufacturing engineers it is doubtless due that the old-time aloofness of the "practical" for the "theoretical" man has so largely been overcome, and it is due in no small degree to the meeting of professor, manufacturer, and student in the common ground of Society fellowship that science and practice have become so closely united.

229 The European trip of 1889 brought many of the members of the Society into close communication with the engineering work of the old world, and even those who had already had the opportunity of examining the workshops and methods of Great Britain and the Continent, found that the unparalleled opportunities afforded during that eventful trip far exceeded anything that could have been acquired by private individuals. The fellowship thus created between the Society and the various European organizations did much to broaden the work of the American association, placing it well in the front as a National society, representing a field of engineering work which had hitherto been relegated to a subordinate position. It is not too much to say that the prominence acquired by the Society in connection with the engineering features of the expositions of 1889 and 1893 gave it a strength and a status which have characterized its movements ever since.

230 Following upon the International Engineering Congress of

the Chicago Exposition came a development of interest in the work of the Society which caused an increase both in its membership and in the character and number of papers and discussions included in its Transactions, a development which will be given a general review in the present chapter, before proceeding with the detailed record of its work.

(To be continued.)

A METHOD OF IMPROVING THE EFFICIENCY OF GAS ENGINES

By Thos. E. Butterfield,[1] Philadelphia, Pa.

Non-Member

The movement toward higher efficiency among gas engine men has been in the direction of lengthening the high end of the indicator card, and securing greater range of expansion by reducing the clearance volume and increasing the compression. Due to the inevitable loss of heat when gases at high temperatures pass through ports and valves, the compound engine has failed to show any gain.

2 The old slide valve could not be used with high pressures, and the replacement of the slide valve by the poppet valve was followed by an increase in compression until a new limitation was reached. The speed of combustion became excessively great, resulting in shocks and loud poundings. Premature ignition or self ignition of the mixture of gas and air was encountered. These troubles caused the practical compression limit to be at four or five atmospheres.

3 With liquid hydrocarbons these troubles were most acute, and Diesel, while searching for the isothermal combustion engine, came upon the principle of controlled combustion and separate compression of gas and air. This cycle makes possible compressions of forty atmospheres but leaves a large part of the intermediate range still unused, because oil injected at the end of the compression will not promptly become gas and burn with much lower compression.

4 Banki cooled the charge and also the surface of the combustion chamber in the Otto cycle engine by introducing water vapor with the vapor of liquid fuel, and he used as high as fifteen atmospheres of compression. Water injection had been used before but not systematically. Banki's engine never attained commercial success. The destructive shocks resulting from accidental interferences with the water supply and the undue corrosion of the valves and the interior

[1] **Chief Engineer Otto Gas Engine Works, Philadelphia, Pa.**

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of the cylinder due to imperfect vaporization of the water were probably the principal causes that prevented extensive use of this engine.

5 Diluting the gaseous mixture with air has been suggested and tried out lately by Mr. Dugald Clerk. The discussion of this proposal recalls an idea first thought of about 20 years ago by Mr. John Saltar, Jr., late president of the Otto Gas Engine Works, and at that time their Chicago agent. It concerned a method of using high compression in the Otto cycle engine by diluting with an inert gas the combustible charge drawn in during the suction stroke. A patent application was drawn up but never submitted; and the following extracts from this application show clearly the essential features in the words of the inventor.

My invention relates to engines wherein individual charges of fluid fuel and air are successively mixed and ignited. It is well known that the efficiency of combustion in engines of this class is increased with the degree of compression at which the charge is ignited. However, such preliminary compression is subject to practical limitation in that a great increase thereof beyond such limitation yields an indicator card showing relatively too great initial pressure and high temperature of combustion, and is accompanied by undue shocks upon the engine, manifested by noisy poundings.

I have ascertained by practical tests that with a fuel which is poorer than gasoline vapor a higher degree of compression may be employed without the disadvantages above specified. So that a relatively poor gas has a higher power-producing value, in an engine of this class, than what may be considered its heating equivalent in a richer gas. For instance, I find that a volume of 4 cu. ft. of ordinary "producer gas," containing 130 B.t.u. per cu. ft. equals in the amount of horse power generated under its proper compression and combustion, a volume of 1 cu. ft. of ordinary "City" gas, containing 650 B.t.u. per cu. ft., or 520 B.t.u. in producer gas, yields as much power as 650 B.t.u. in City gas. Moreover, I find that if a charge of gasoline vapor and air be modified by the addition and proper admixture therewith of a percentage of inert fixed gas (i.e., one which neither burns nor supports combustion), the charge can be subjected to a higher preliminary compression than would be practical for that particular fuel in its undiluted condition; the initial pressure of the combustion being reduced without a proportionate reduction of the mean effective pressure; and a large saving is effected in the total quantity of gasoline required to produce a given result in horse power.

I have found in practice, that the waste inert products of previous combustion, by proper treatment, may be economically utilized in diluting the charge of rich fuel, and, as I find it desirable to eliminate the moisture of combustion and also to reduce the temperature of the products of combustion, for this purpose, the embodiment of my invention selected for illustration comprises apparatus to accomplish such result.

I have operated such an engine with charges thus diluted; igniting them at a pressure of 120 lb. per sq. in., and, without disadvantageous poundings or exces-

sive initial pressure and temperatures, the noise of combustion being practically no greater than that incident to the ignition of the undiluted mixture of gasoline vapor and air at a pressure of 60 lb. per sq. in. The result of such operation was the saving of 25 per cent of the gasoline, in producing a given horse power.

Although I find it convenient so to arrange my improved engine that its supply of diluting material is obtained from its own exhaust, it is to be understood that any fixed gas which is capable of thorough admixture with the charge of fuel and air without imparting undesirable qualities thereto, and which is so substantially inert in comparison with the fuel itself as to diminish the heat value thereof per unit of charge, may be employed for the purpose described.

6 Experiments designed to show the practical working effect of this method were made in Chicago, prior to 1898, and at The Otto Gas Engine Works, Philadelphia, 1899–1902.

7 The first experiments at Philadelphia were made in October and November 1899. A 20 h.p. Otto gasolene engine, with cut-out governor and automatic air valve, was used. Its bore was $8\frac{1}{2}$ in.; stroke 15 in.; revolutions 240 per minute; stroke volume $851\frac{2}{10}$ cu. in.; compression space $26\frac{36}{100}$ per cent of the stroke volume.

8 The dilution of the charge with exhaust gas was first effected by lengthening the exhaust cam, thus holding the exhaust valve open during a portion of the suction stroke and drawing in exhaust gas from the exhaust pipe while the gasolene vapor and air were drawn in through the inlet valve. Tests were made successively with the exhaust valve closing at 86 deg., 73 deg., 60 deg., 41 deg., and 25 deg., after the beginning of the suction stroke. The best results were obtained with the exhaust valve opening at 127 deg. after the beginning of the explosion stroke, and closing at 25 deg. after the beginning of the suction stroke. With a compression pressure of 80 lb. per sq. in. a fuel consumption of $\frac{8}{10}$ pints of gasolene per brake horse-power hour was obtained with quiet running. The maximum brake horse-power however, was reduced from 21 to 19.

9 With the exhaust valve closing early and thus retaining a larger than normal proportion of exhaust gas in the cylinder, the efficiency was reduced and the explosions were much more violent. This was no doubt due to retention of an excessive amount of heat in the cylinder.

10 To insure a cool charge and a uniform supply of burnt gas, in another experiment, the exhaust of a smaller engine was led through two exhaust vessels and then carried to the air inlet holes in the frame of the engine. This gave a better efficiency, a fuel consumption of 0.75 pints per brake horse-power per hour being obtained with smooth running.

11 This compression was very moderate and not thoroughly adapted to test the influence of burnt gas dilution in making the running quiet.

12 In order to separate the gasolene and air further and thus reduce the suddenness of the explosion, burnt gas instead of air was used to vaporize the oil, but the effect of the change in method was but slight.

13 Another series of experiments was made in February and March 1902 with a 15 h.p. Otto gasoline engine with 6¾ in. bore, 15½ in. stroke and 260 r.p.m. The compression space was 17 per cent of the stroke volume. The burnt gas inlet was controlled by a timed poppet valve. Several experiments were made with short connections between the exhaust and inlet, but the best results were obtained with a long pipe connection containing about 9 stroke-volumes of burnt gas. The maximum brake horse-power was about 14, and the gasolene consumption per brake horse power per hour in pints 0.692, 0.701, 0.710, 0.691, 0.702, 0.697, 0.691, 0.69. The pounding was a little louder than with the low compression engine, but could be kept within permissible limits.

14 An attempt to follow the action of inert gas dilution by means of figures was made at the time of the later experiments. As a basis for calculating the results given in the table, it was necessary to make certain assumptions which cannot be verified. Nevertheless, any error in any of these figures will not affect the usefulness of the comparison drawn from the table.

15 The temperature of the burnt gas in the clearance at the beginning of the suction stroke is assumed at 1200 deg. fahr., absolute. This is, of course, much lower than the temperature at the beginning of the exhaust stroke.

16 The temperature of the mixture as it enters the cylinder is taken at 530 deg. fahr., absolute. This varies of course, with different liquid fuels, according to the vapor tension and the latent heat of vaporization; for instance, grain alcohol with a latent heat of 371 B.t.u. per pound, will, if perfectly vaporized, lower the temperature of the explosive mixture 148 deg. fahr., and wood alcohol with a latent heat of 481 B.t.u. per lb. will cause a drop of 258 deg. fahr. Such fuels with high latent heat of vaporization are only partially evaporated, and there is probably a considerable proportion of such fuels present in the liquid form at the time of ignition, either as a mist or as condensation on the cooler parts of the walls of the combustion chamber. The tremendous absorption of heat by a liquid

so diffused through the charge at the time of inflammation probably has an important influence in reducing the speed of combustion and thus making possible high compression with these fuels.

17 Atmospheric pressure is taken at 15 lb. and the pressure in the cylinder at the end of the suction stroke is taken at 14 lb. per sq. in. absolute.

18 The proportion of burnt gas in the entering charge is P, and the percentage will be 100 P. The ratio of clearance to stroke-volume is C. The temperature of mixed gases in the cylinder at the end of the suction stroke is T_1. The specific heat of the charge before and after burning is taken the same; 0.17 at constant volume and 0.22 at constant pressure, with the ratio 1.3.

19 With a cylinder stroke-volume of one cubic foot, the burnt gas left in the cylinder at the end of the suction stroke expands to $\frac{15}{14}$ C, the exhaust stroke being finished at atmospheric pressure.

20 Assuming no passage of heat to the cylinder walls, the heat lost by this burnt gas left in the clearance in mixing is the same as the heat gained by the entering charge. And as the thermal capacities of equal volumes of permanent gases reduced to the same pressure and temperature are always the same, the equation follows:

$$\frac{15}{14}C \times \frac{492}{1200}(1200 - T_1) = \left(1 + C - \frac{15}{14}C\right) \times \frac{492}{530}(T_1 - 530)$$

$$T_1 = \frac{1 + C}{\left(1 + C - \frac{15}{14}C\right)\frac{1}{530} + \frac{15\,C}{14 \times 1200}} = \frac{530\,(1 + C)}{1 + \frac{4}{10}C} \text{ Approx.}$$

The compression pressure is given by the formula

$$P = 14\left(\frac{(1 + C)}{C}\right)^r$$

The temperature of compression is

$$T^2 = T\left(\frac{1 + C}{C}\right)^{r-1} = \left(\frac{1 + C}{C}\right)^{\frac{3}{10}}$$

21 The number of B.t.u. released by combustion is taken at 52$\frac{1}{4}$ for each cubic foot of the perfect combustible mixture of fuel and air. This low value gives figures for the explosion pressure somewhere near the actual, without introducing any figures in regard to suppressed combustion or possible variation of specific heat. The num-

ber of B.t.u. per cu. ft. stroke-volume of the mixture actually present in the cylinder at the end of the suction stroke is

$$52\tfrac{1}{4} \times \frac{492}{530}(1 - P) \times \frac{14 - C}{15} = 3.234\ (1 - P)\ (14 - C)$$

The number of B.t.u. per cubic foot of clearance volume is

$$3.234\ \frac{(1 - P)\ (14 - C)}{C}$$

22 The approximate composition of burnt gas when freed from water is taken at about 18½ per cent carbonic acid gas, 2½ per cent of unburnt excess oxygen, and 79 per cent of nitrogen. A cubic foot of this mixture will weigh about 0.0868 lb. at 32 deg. fahr. A cubic foot of air weighs about 0.0808 lb. and when carburetted with heavy gasolene vapor will weigh about 0.0837 lb. per cubic foot.

23 Adding the weight of burnt gas left in the clearance, which is equal to

$$0.0868\ C\ \frac{492}{1200} = 0.03559\ C$$

to the weight of mixture drawn in during the suction stroke, which is

$$0.0837 \cdot \frac{14 - C}{15} \cdot \frac{492}{530}$$

the total weight of charge in the cylinder at the end of compression is, approximately, $0.0725 + 0.0304\ C$ and the thermal capacity of this charge is $0.17\ (0.0725 + 0.0304\ C) = 0.01233 + 0.0052\ C$.

24 The rise in temperature of the charge on complete combustion is

$$\frac{3.234\ (1 - P)\ (1 - C)}{0.01233 - 0.0052\ C}$$

25 The thermal efficiency is

$$E = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{C}{1 + C}\right)^{\frac{3}{10}}$$

and the mean effective pressure is

$$\frac{778\ \mathrm{E} \times \text{B.t.u. per cu. ft.}}{144}$$

Stroke-volume = $0.54\ EH$.

26 The figures for a moderate compression of 30 per cent of stroke volume and a very high compression of 15 per cent of the stroke volume, are given below for easy reference:

Clearance per cent	% Rise	30	15
Compression temperature	11	952	1060
Compression pressure	111	94	198
Rise in temperature at explosion	7	3188	3410
Explosion temperature	8	4140	4470
Explosion pressure	104	409	834
Mean effective pressure	29	85	110
Thermal efficiency	29	35.6	45.7
Ratio weight explosive mixture to weight burnt gas.	103	6.55	13.28

27 This shows the rise in explosion temperature to be only 8 per cent, which is inconsiderable, but both compression and explosion pressures rise very rapidly, and the dilution of the charge by burnt gases left in the clearance space is reduced more than 50 per cent.

28 The actual pressure and temperatures in the cylinder at the end of compression can have no important influence on the character of the explosion, for with 30 per cent clearance it is necessary only to burn a little more than 3 per cent of the charge before the temperature in the cylinder is higher than that at the end of compression with 15 per cent clearance; and, similarly, it is necessary only to burn one-third of the charge with 30 per cent clearance before the pressure in the cylinder rises to the same figure as at end of compression with 15 per cent clearance. The remaining portion of the charge is not burnt at an increasing rate of speed, but rather at a decreasing rate, even in the presence of these high pressures and temperatures.

29 To explain the difference in behavior of the highly compressed charge, we are accordingly compelled to neglect compression temperatures and pressures, leaving for consideration the difference in density of the compressed charges as expressed in heat units per cubic foot and in pounds per cubic foot.

30 With the higher compression, the particles or molecules of gas and oxygen are shoved closer together, the intermolecular distances with 30 per cent and 15 per cent clearance roughly comparing as the cube root of two to one. As combustion at constant volume proceeds, particles or molecules of carbonic acid are being constantly interposed between the active molecules of gas and oxygen. It is true that the condensible hydrocarbons probably do not burn directly to carbonic acid, but may pass through one or more intermediate stages, but the action even in this case is similar, in

that the most active elements become further separated from the instant when combustion starts.

31 The speed of combustion which we know to obtain in gaseous mixtures is probably increased by these intervening particles of more or less inert gas acting as carriers The process may be said to begin with the gas and oxygen molecules adjacent everywhere throughout the charge, except for the separation by nitrogen. As the combustion proceeds the particles of CO_2,CO, the aldehyds, etc., are pushed in between, making the path of transfer between the active particles longer every instant.

32 When the process of combustion has gone far enough to place a certain critical amount of inactive gas between the particles of active gas, the speed of burning rapidly falls off, and this assumed phenomenon furnishes a rational explanation of the much discussed "suppressed combustion." This occurs even with such gases as CO and H, where the formation of an intermediate compound is practically barred, and it also always occurs in the face of the approach to maximum pressures and temperatures.

33 Since suppressed combustion of gaseous mixtures is present where low compression and explosion pressures and temperatures exist, and also where high compression and explosion pressures and temperatures exist, this action must be dependent on some relation or cause not dependent on the variations induced by change in clearance, such as in pressures, temperatures and densities.

34 The gradual dilution of the charge by the more or less inert products of its own complete or partial combustion is, however, always a feature of such combustion, and may reasonably be supposed to be the determining factor in fixing the character of this action.

35 If this automatic dilution has such a marked influence on the nature of every combustion or "explosion" that takes place in the cylinder of a gas engine, it is beyond doubt that the artificial creation of such conditions will also result in the production of corresponding effects. In other words, dilution of the charge with burnt gas, or perhaps with other inert gas, will result in the creation of suppressed combustion at the beginning of combustion, instead of as in the undiluted charge, waiting until combustion is partially complete. By varying the amount of such initial dilution, also, the character of the combustion may be controlled within wide limits.

36 Especially in small engines, a practical difficulty presents itself in the increasing difficulty of securing positive ignition as the quality of the charge is altered by dilution.

37 With a strong electric spark, ignition is more prompt and certain wth high compression and consequently poorer charges may be certainly ignited; but the dilution of the charge necessary to secure slow combustion with high compression is probably so great that certain ignition cannot be relied upon under all conditions. In larger engines the difficulty is not likely to be so serious, and by using a small portion of the total charge without any dilution, the ignition may always be made certain.

38 By advancing the time at which suppressed combustion becomes evident, and lengthening its extent, the dilution of the charge with inert gas causes a diminution of the maximum temperature range with any compression, and thus involves a loss in efficiency. But with a practical and reasonable amount of dilution, or just enough to make the use of a high compression possible, this loss is bound to be inconsiderable compared to the gain resulting from the use of the higher compression.

39 The chief drawbacks to the use of high compression with dilution are the loss in mean effective pressure, and the danger of destructive shocks should undiluted charges be accidentally drawn into the cylinder and compressed and ignited.

40 In the tests made at the shops of The Otto Gas Engine Works no meters were used to determine the exact proportions of the various constituents of the mixture drawn into the cylinder. The percentage of burnt gas used was estimated from the loss in mean effective pressure, a method open to serious objection. The effort was made to separate out all the water of condensation in the burnt gas created by the combustion of hydrogen, but no hygrometric tests were made to determine the exact quantity of water present. As the amount of aqueous vapor in a charge may have a quite appreciable effect on the character of the combustion, the absence of such a test was a drawback.

41 Refined measurements are seldom possible in ordinary commercial work, however, especially when exploring a new field; and it is to be hoped that the subject may be exhaustively investigated in some of our college laboratories.

TABLE 1 THERMAL DATA FOR NO DILUTION OF CHARGE

ABSOLUTE TEMPERATURES AND PRESSURES

Clearance in per cent of Stroke Volume	35	32	30	27	25	22	20	17	15	12	10
Temperature at End of Suction	628	620	613	608	604	594	588	582	575	566	561
Pressure of Compression	81	88	94	100	113	130	144	172	198	255	316
Temperature of Compression	941	948	952	967	979	993	1006	1039	1060	1106	1152
B.t.u. per cu. ft. Stroke-volume	44.1	44.2	44.3	44.4	44.4	44.5	44.6	44.2	44.7	44.8	44.9
B.t.u. per cu. ft. of Clearance	126	134	147	164	178	212	223	263	298	373	449
Maximum Temperature	4060	4110	4140	4200	4235	4290	4310	4420	4470	4550	4640
Maximum Pressure	349	381	409	434	488	552	616	732	834	1050	1255
Thermal Efficiency	33.2	34.6	35.6	37.1	38.3	40.2	41.6	44.0	45.7	48.8	51.3
Mean Effective Pressure	79.0	82.4	85.0	89.0	91.5	96.5	100	106	110	118	124
Total Weight of Gas per cu. ft. Stroke-volume	0.0831	0.0822	0.0816	0.0807	0.0801	0.0792	0.0786	0.0777	0.0771	0.0762	0.0756
Weight Burnt Gas per cu. ft. Stroke-vol.	0.0126	0.0115	0.0108	0.0097	0.0090	0.0079	0.0072	0.0061	0.0054	0.0043	0.0036
Ratio Weight of Explosive Mixture to Burnt Gas	5.59	6.13	6.55	7.32	7.90	9.03	9.91	11.75	13.28	16.70	20.00
Weight Explosive Mixture per cu. ft. Stroke-volume	0.0705	0.0707	0.0708	0.0710	0.0711	0.0713	0.0714	0.0716	0.0717	0.0719	0.0720

TABLE 2 EFFECT OF DILUTION OF CHARGE WITH INERT GAS

CLEARANCE 15 PER CENT OF STROKE-VOLUME

Per cent of Inert Gas in Entering Charge	None	10	15	20	25	30	35	40	45	50
Per Cent of Inert Gas in Total Volume	7.2	16.5	21.1	25.8	30.4	35.0	39.7	44.3	49.0	53.6
B.t.u. per cu. ft. of Stroke-Volume	44.7	40.2	38.0	35.8	33.5	31.3	29.1	26.8	24.6	22.3
B.t.u. per cu. ft. of Clearance	298	268	253	238	224	209	194	179	164	149
B.t.u. per Pound	580	520	491	461	431	402	372	343	314	285
Rise in Temperature at Explosion	3410	3055	2885	2708	2534	2363	2189	2019	1844	1674
Maximum Temperature	4470	4115	3945	3768	3594	3423	3249	3079	2904	2734
Maximum Pressure	834	768	737	703	671	640	608	576	543	511
Mean Effective Pressure	110	99	93	88	83	77	71	66	61	55

SAFETY VALVE MEETING

SAFETY VALVES

By Frederic M. Whyte, New York
Member of the Society

The general subject of safety valves is such a very broad one that it would be impossible to consider it fully in one paper and it will be my purpose merely to open the subject of safety valves for steam boilers for discussion. In general the engineering profession has been quite ignorant about the principles of safety valves, their relative capacities, and the capacities required for various conditions of steam generation; inasmuch, therefore, as these items have recently been given very extended and careful study, and many interesting data have been collected, this may be considered an appropriate time and place to make such information available. It is the purpose of this paper to present some ideas about safety valves for steam boilers and particularly for locomotive boilers.

2 Just how the capacity of the first valve used on a steam boiler was determined, or what relation this capacity had to the generating capacity of the boiler, may be recorded somewhere in history, but it is doubtful if either fact was recorded or even determined. So far as locomotive work is concerned, the same ignorance prevails today; it is well to remember, however, that there is good promise that this ignorance will soon be dispelled. In marine work certain formulae have been devised for calculating the sizes of safety valves, and these formulae have been accepted, more or less blindly, it is thought.

3 As the writer is more familiar with locomotive work than with other lines, it will be more consistent to confine these remarks to that branch of the subject, with the understanding, however, that those who discuss the subject may consider locomotive, marine and stationary boilers.

Presented at the monthly meeting (February 1909) of The American Society of Mechanical Engineers. All papers are subject to revision.

PRACTICE IN LOCOMOTIVE WORK

4 The general practice in locomotive work has been to determine in an "offhand" way the size and number of safety valves to be used, and former practice has guided the determination entirely. If a larger boiler is to be used the valve capacity may not be increased, depending upon the judgment of the person whose duty it is to determine the capacity. Again, the capacity has been indicated in an indifferent manner, being expressed as a "size," meaning the diameter of something more or less uncertain; while the other dimension, the lift, which is necessary to give an indication of the capacity, is entirely ignored.

5 But to know the exact capacity of the available valves is not sufficient; it is quite as important to know how much steam is to be released and in what length of time it should be released. It will be comparatively easy to determine the capacity of safety valves, if indeed the elaborate tests which have already been made—data from which it is hoped may be presented in the discussion—have not already solved part of the problem; more difficult will be that part of it which is concerned with the quantity of steam to be released and the rate of the release. The subject is of mutual interest to the valve manufacturer and the user, the design of the valve for capacity and wear to be worked out by the manufacturer, the capacity which is to be used, both in volume and in number of valves, and the rate of release, to be determined by the user with the assistance of the manufacturer.

ESSENTIALS OF A SAFETY VALVE ON A LOCOMOTIVE

6 The design of the valve will include the diameter of the controlling opening and the passages leading to it from the steam volume, as well as those leading from it to the atmosphere, the shape and material of the seat, the amount of lift of the valve, and the shape and material of the valve face, the spring and its protection, the adjustment, the muffler, if one is to be used, and the action of the valve in lifting and in seating.

7 It will not be necessary to discuss the diameter of the controlling opening, and of the passages to and from it, in view of the suggestion here made that instead of indicating the capacity of a valve in a very rough way by the diameter of some opening, the capacity be expressed in pounds of steam at certain pressures. The shape of the seat and of the valve face may or may not be of importance; but this

will be referred to again. The material in the seat and face will naturally be that which will best withstand the effects of the flow of steam over them, and the possible pound of the valve when seating.

8 The reliability of the spring and the effect of heat upon it are very important points. Adjustments should be readily made, but on the other hand to get out of adjustment should be practically impossible. The capacity of the muffler need not be questioned, except in extreme designs, but the indicated capacity should be that of the valve complete, with or without muffler, according to the intended use of the valve; then it is important only that it deaden sufficiently the noise of the escaping steam.

9 The action of the valve in lifting and in seating, the desirability of a forewarning that the maximum pressure is about reached, and the operating conditions which bear upon this question of forewarning, are correlative. With any kind of steam-generating plant it ought to be quite sufficient if those immediately responsible for the quantity produced, and for its use, know what is available; in stationary and in marine work this is generally true, and steam gages can be placed within view of those who should know what the pressure is at any time; unfortunately in locomotive work, however, it has become perhaps desirable that others than those within view of the gage in the cab know something about the steam pressure, and inasmuch as the fireman is willing, and sometimes anxious, that they should know, he takes the only means at hand to inform them when he thinks that the results of his labors are good, and "fires against the pop" so that everybody within hearing or sight of the valve knows by the escaping steam that the fireman is doing his duty. If when a train is ascending a grade the conductor at the rear sees steam escaping from the valve he knows the train will get up the grade; on hard grades he will watch for the only indication which can be given him, and the fireman tries to present this indicator, the escape from the valve, the "white feather."

10 Numerous similar examples might be mentioned, but assuming that such an indicator of steaming conditions has grown to be a necessity, undesirable as it may be, how can it be produced at the least expense? Surely not with a valve from 2½ to 4 in. in diameter and open to its full capacity. Two devices, at least, are available, to give the indication at a lower cost than the full open valve: the "simmering" valve, which will open slightly for two or three pounds, about the normal maximum, and then open full, just reversing this in seating; the other, the small pilot valve, which will open at two or

three pounds pressure below the working valve. The first method will have some bearing on the kind of metal to be used in the valve seat and valve face and possibly upon the shape of the exterior edge of the valve and the opposing surface of the seat. The second method means the addition of the small valve, an additional cost for which there will be no need if the first method can be developed successfully.

RELATION OF VALVE CAPACITY TO STEAM-GENERATING CAPACITY

11 There remains for consideration the relation of valve capacity to steam-generating capacity, and the unit capacity of the valves which will make up the total valve capacity. The fact that in locomotive work the total valve capacity has not been as great as the maximum steam-generating capacity should be ample proof that such valve capacity is not necessary. The reason for this is, of course, that on account of using the exhaust steam for producing the forced draft, when the demand for steam from the boiler is reduced or entirely cut off, the forced draft is automatically reduced or cut off, and the generating capacity is reduced so that it is not necessary that the safety valves release the full generating capacity. Probably it will be largely a question of opinion what per cent of the total generating capacity the valve ought to have, although it is possible that as attention is centered upon this question some more or less positive solution of it may result.

12 Having fixed upon the per cent of the generating capacity to be provided for in the valves it will be necessary to determine the desirable unit capacity of the valves. Some states require that each locomotive boiler have at least two valves. Starting with this condition, consideration of the maintenance of the valves indicates that they should be duplicates and therefore that each have a capacity equal to one-half the required generating capacity. If a number of boilers of different capacities are to be considered, then the smaller ones would probably be provided with the same valves as the larger ones for the purpose of duplication. There are some large boilers for which three valves might be necessary, because the necessary capacity in two units might make the valves abnormally large for construction purposes. Also it is worth while to consider whether undesirable results would come about from opening almost instantaneously an escape of steam from the boiler to the atmosphere. No suggestions are offered on this, but it is hoped that something bearing on the question may be developed in the discussion.

13 It is suggested that instead of setting the several valves on a boiler at different pressures, all the valves be set at the same pressure, with the idea that each of them will operate frequently enough to keep all in working condition, rather than run the risk of one valve being found out of condition when it is required for action.

14 It is probable that some time it will be found desirable to consider the minimum distance above the steam releasing surface of the water at which the safety valve seat may be placed.

SAFETY VALVE CAPACITY

By Philip G. Darling, New York

Associate Member of the Society

The function of a safety valve is to prevent the pressure in the boiler to which it is applied from rising above a definite point, to do this automatically and under the most severe conditions which can arise in service. For this, the valve or valves must have a relieving capacity at least equal to the boiler evaporation under these conditions. If it has not this capacity, the boiler pressure will continue to rise, although the valve is blowing, with a strain to the boiler and danger of explosion consequent to over-pressure. Thus, with the exception of the requisite mechanical reliability, the factor in a safety valve bearing the most vital relation to its real safety is its capacity.

2 It is the purpose of this paper to show an apparatus and method employed to determine safety valve lifts, giving the results of tests made with this apparatus upon different valves; to analyze a few of the existing rules or statutes governing valve size; and to propose a rule, giving the results of a series of direct capacity tests upon which it is based; to indicate its application to special requirements; and finally its general bearing upon valve specifications.

3 Two factors in a safety valve geometrically determine the area of discharge and hence the relieving capacity:—the diameter of the inlet opening at the seat and the valve lift. The former is the nominal valve size, the latter is the amount the valve disc lifts vertically from the seat when in action. In calculating the size valves to be placed on boilers, rules which do not include a term for this valve lift, or an equivalent, such as a term for the *effective* area of discharge, assume in their derivation a lift for each size valve. Nearly all existing rules and formulae are of this kind, which rate all valves of a given nominal size as of the same capacity.

4 To find what lifts standard make valves actually have in prac-

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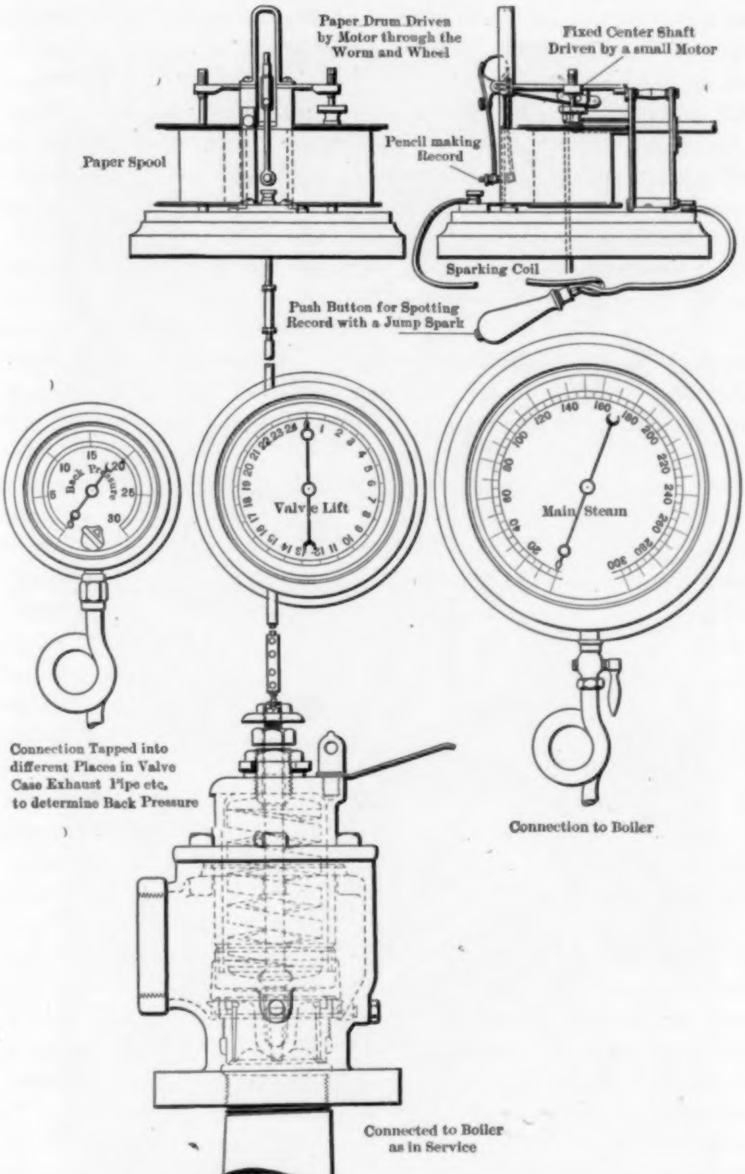

FIG. 1 SAFETY VALVE LIFT RECORDING APPARATUS

tice, and thus test the truth or error of this assumption that they are approximately the same for the same size valve, an apparatus has been devised and tests conducted upon different makes of valves. With this apparatus not only can the valve lift be read at any moment to thousandths of an inch, but an exact permanent record of the lift during the blowing of the valve is obtained, somewhat similar to a steam engine indicator card in appearance and of a quite similar use and value in analyzing the action of the valve.

5 As appears in Fig. 1 the valve under test is mounted upon the boiler in the regular manner, and a small rod is tapped into the top end of its spindle, which rod connects the lifting parts of the valve directly with a circular micrometer gage, the reading hand of which indicates the lift upon a large circular scale or dial. The rod through this gage case is solid, maintaining a direct connection to the pencil movement of the recording gage above. This gage is a modified Edson recording gage with a multiplication in the pencil movement of about 8 to 1, and with the chart drum driven by an electric motor of different speeds, giving a horizontal time element to the record. The steam pressures are noted and read from a large test gage graduated in pounds per square inch, and an electric spark device makes it possible to spot the chart at any moment, which is done as the different pound pressures during the blowing of the valve are reached. The actual lift equivalents of the pencil heights upon the chart are carefully calibrated so that the record may be accurately measured to thousandths of an inch.

6 In testing, the motor driving the paper drum is started and the pressure in the boiler raised. The valve, being mounted directly upon the boiler, then pops, blows down and closes under the exact conditions of service, the pencil recording on the chart the history of its action.

7 With this apparatus, investigations and tests were started upon seven different makes of 4-in. stationary safety valves, followed by similar tests upon nine makes of muffler locomotive valves, six of which were $3\frac{1}{2}$ in., all of the valves being designed for and tested at 200 lb. The stationary valve tests were made upon a 94-h.p. water-tube boiler made by the Babcock & Wilcox Co. (See Fig. 2.) The locomotive valve tests were made upon locomotive No. 900 of the Illinois Central R. R., the valve being mounted directly upon the top of the main steam dome. (See Fig. 4 and 5.) This locomotive is a consolidation type, having 50 sq. ft. of grate area and 2953 sq. ft. of heating surface. Although a great amount of addi-

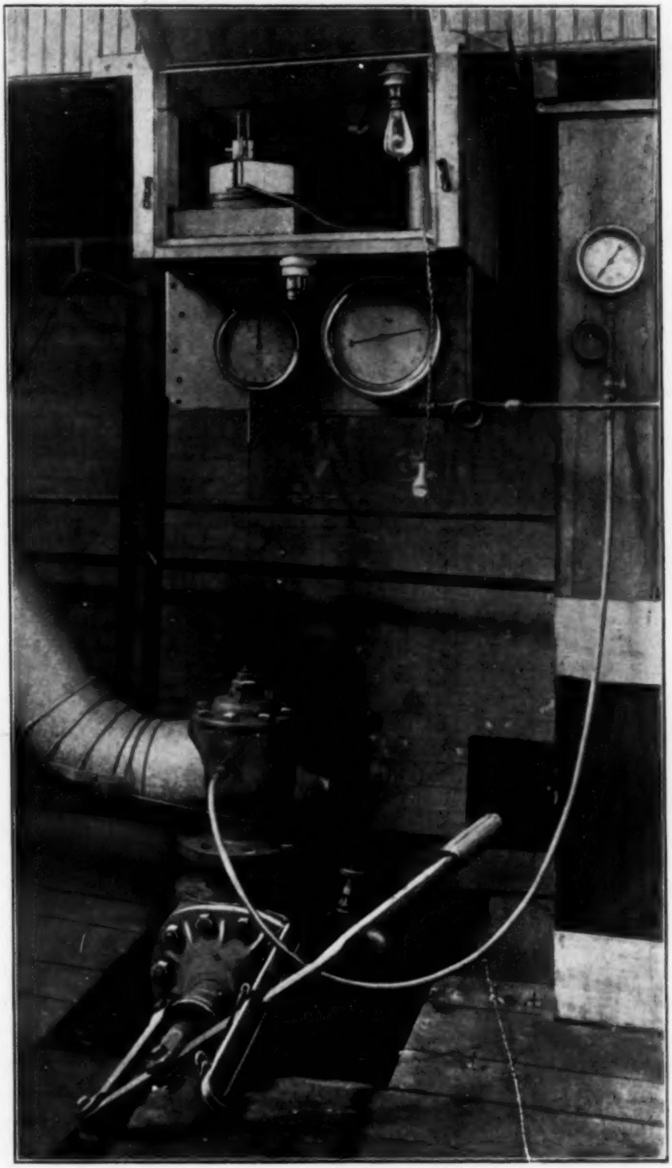

FIG. 2 VALVE LIFT APPARATUS AS USED WITH THE STATIONARY TEST BOILER AT BRIDGEPORT, CONN.

tional experimenting has been done, only the results of the above tests will be quoted here. These lift records show (with the exception of a small preliminary simmer, which some of the valves have) an abrupt opening to full lift and an almost equally abrupt closing when a certain lower lift is reached. Both the opening and closing lifts are significant of the action of the valves. (See Fig. 3.)

8 The results of the 4-in. iron body stationary valve tests summarized are as follows: of the seven valves the average lift at opening was 0.079 in. and at closing 0.044 in., or excluding the valve with the highest lifts, the averages were 0.07 in. at opening and 0.037 in. at closing. The valve with the lowest lifts had 0.031 in. at opening and 0.017 in. at closing, while that with the highest had 0.137 in. and 0.088 in. Expressing the effective steam-discharge areas of the

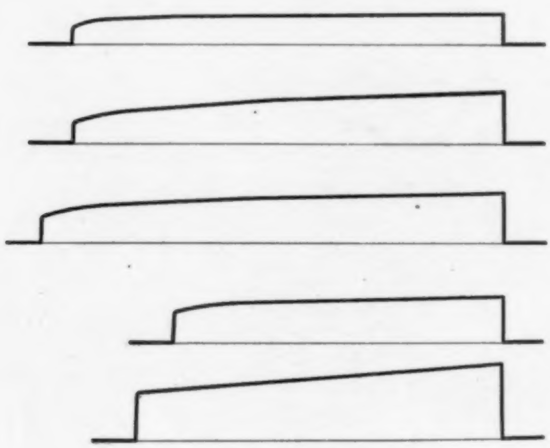

Fig. 3 Typical Valve-Lift Diagrams

valves taken at their opening lifts as percentages of the largest obtained, the smallest had 31.4 per cent, the next larger 40.8 per cent, and the next 46.6 per cent. Of the six 3½-in. muffler locomotive valves the summarized lifts are as follows: average of the six valves, 0.074 in. at opening and 0.043 in. at closing. Average excluding the highest, 0.061 in. at opening and 0.031 in. at closing. The lowest lift valve had 0.04 in. opening and 0.023 in. closing; the highest, 0.140 in. opening and 0.102 in. closing. As percentages of the largest effective steam-discharge area, the smallest was 36.4 per cent, the next larger 39.8 per cent, and the next 46.4 per cent. In both the

stationary and locomotive tests, the lowest lift valve was flat-seated, which is allowed for in the above discharge area percentages.

9 The great variation—300 per cent—in the lifts of these standard valves of the same size is startling and its real significance is apparent when it is realized that under existing official safety valve rules these valves, some of them with less than one-third the lift and capacity of others, receive the *same* rating and are listed as of equal relieving value. Three of these existing rules are given as an illustration of their nature; the United States Supervising Inspectors

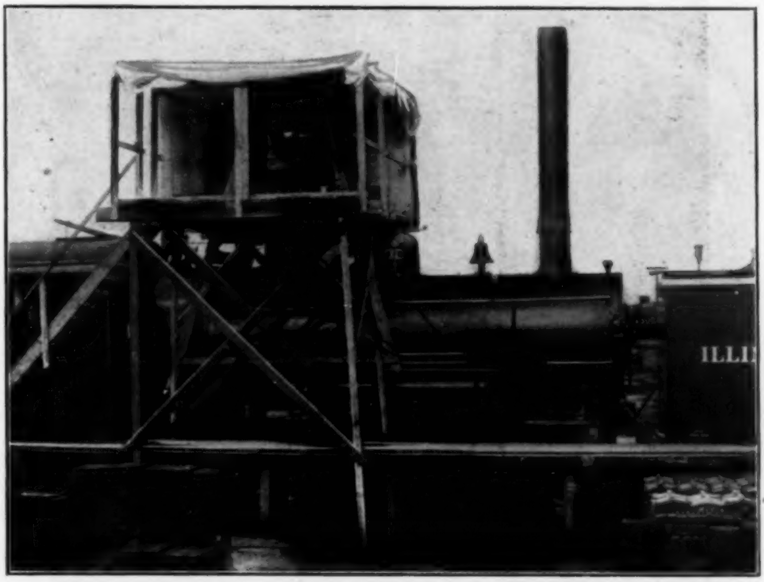

Fig. 4 Valve Lift Apparatus as Erected for Locomotive Testing at Burnside, Ill.

Rule, the Boiler Inspection Rule of Philadelphia and that of the Board of Boiler Rules of Massachusetts.

RULE OF UNITED STATES BOARD OF SUPERVISING INSPECTORS

$$A = 0.2074 \times \frac{W}{P}$$

A = area of safety valve in square inches per square foot of grate surface.
W = pounds of water evaporation per square foot of grate per hour.
P = boiler pressure (absolute).

10 In 1875 a special committee was appointed by the United States Board of Supervising Inspectors to conduct experiments upon safety valves at the Washington Navy Yard. Although the pressures used in these experiments (30 x 70 lb. per square inch) were too low to make the results of much value today, one of the conclusions reported is significant:

Fig. 5 Detail of Lift Apparatus at Burnside, Ill., Showing Locomotive Valve

a That the diameter of a safety valve is not an infallible test of its efficiency.

b That the lift which can be obtained in a safety valve, other conditions being equal, *is* a test of its efficiency.

11 The present rule of the board, as given above, formulated by Mr. L. D. Lovekin, Chief Engineer of the New York Shipbuilding Co., was adopted in 1904. Its derivation assumes practically a 45-deg. seat and a valve lift of $\frac{1}{32}$ of the nominal valve diameter. The discharge area in this rule is obtained by multiplying the valve lift $\frac{D}{32}$ by the valve circumference ($\pi \times D$) and taking but 75 per cent of the result to allow for the added restriction of a 45-deg. over a flat seat. The 75 per cent equals approximately the sine of 45 deg. or 0.707. This value for the discharge area $\left(\text{i.e., } 0.75 \times \pi \times \frac{D^2}{32}\right)$ is substituted directly into Napier's formula for the flow of steam, $w = a\ \frac{P}{70}$. Thus in the valves to which this rule is applied the following lifts are assumed to exist: 1-in. valve, 0.03-in.; 2-in. valve, 0.06-in.; 3-in. valve, 0.09 in.; 4-in. valve, 0.13 in.; 5-in. valve, 0.16 in.; 6-in. valve, 0.19 in.

12 Referring back to the valve lifts given in Par. 8, it is seen that the highest lift valve agrees very closely with the lift assumed in the rule for 4-in. valves, and if the valve lifts of the different designs were more uniformly of this value or if the rule expressly stipulated either that the lift of $\frac{1}{32}$ of the valve diameter actually be obtained in valves qualifying under it, or that an equivalent discharge area be obtained by the use of larger valves, the rule would apply satisfactorily to that size of valve. However, the lowest lift valve actually has but $\frac{1}{4}$, the next larger less than $\frac{1}{2}$, and the average lift of the valves, excluding only the highest, which average is 0.07 in., is but 56 per cent of the lift assumed in the rule for these 4-in. valves.

MASSACHUSETTS RULE OF 1909

$$A = \frac{w \times 70}{P} \times 11$$

A = area of safety valve in square inches per sq. ft. of grate surface.

w = pounds of water evaporation per square feet of grate surface per second.

P = boiler pressure (absolute) at which valve is set to blow.

13 One of the most recently issued rules is that contained in the pamphlet of the new Massachusetts Board of Boiler Rules, dated March 24, 1908. This rule is merely the United States rule given above with a 3.2 per cent larger constant and hence requiring that amount larger valve. The evaporation term is expressed in pounds per second instead of per hour and two constants are given instead of one, but when reduced to the form of the United States rule it gives $A = 0.214 \times \frac{W}{P}$. Figuring this back as was done above with the United States rule, and taking the 75 per cent of the flat seat area as there done, this rule assumes a valve lift of $\frac{1}{33}$ of the valve diameter instead of the $\frac{1}{32}$ of the United States rule. This changing of the assumed lift from $\frac{1}{32}$ to $\frac{1}{33}$ of the valve diameter being the only difference between the two rules, the inadequacy of the United States rule just referred to applies to this more recent rule of the Massachusetts Board.

PHILADELPHIA RULE

$$a = \frac{22.5\,G}{p+8.62}$$

a = total area of safety valve or valves in square inches.
G = grate area in square feet.
p = boiler pressure (gage).

14 The Philadelphia rule now in use came from France in 1868, where it was the official rule at that time, having been adopted and recommended to the city of Philadelphia by a specially appointed committee of the Franklin Institute, although this committee frankly acknowledged in its report that it "had not found the reasoning upon which the rule had been based." The area a of this rule is the effective valve opening, or as stated in the Philadelphia ordinance of July 13, 1868, "the least sectional area for the discharge of steam." Hence if this rule were to be applied as its derivation from the French requires, the lift of the valve must be known and considered whenever it is used. However, the example of its application given in the ordinance as well as that given in the original report of the Franklin Institute committee, which recommended it, shows the area a applied to the nominal valve opening. In the light of its derivation, this method of using it takes as the effective discharge area the valve opening itself, the error of which is very great. Such use, as specifically stated in the report of the committee above referred to,

assumes a valve lift at least $\frac{1}{4}$ of the valve diameter, i.e., the practically impossible lift of 1 in. in a 4-in. valve. Nevertheless, this is exactly the method of use indicated in the text of the ordinance.

15 The principal defect of these rules in the light of the preceding tests is that they assume that valves of the same nominal size have the same capacity and they rate them the same without distinction, in spite of the fact that in actual practice some have but $\frac{1}{3}$ of the capacity of others. There are other defects, as have been shown, such as varying the assumed lift as the valve diameter, while in reality with a given design the lifts are more nearly the same in the different sizes, not varying nearly as rapidly as the diameters. And further than this, the lifts assumed for the larger valves are nearly double the actual average obtained in practice.

16 The direct conclusion is this, that existing rules and statutes are not safe to follow. Some of these rules in use were formulated before, and have not been modified since, spring safety valves were invented, and at a time when 120 lb. was considered high pressure. None of these rules takes account of the different lifts which exist in the different makes of valves of the same nominal size, and they thus rate exactly alike valves which actually vary in lift and relieving capacity over 300 per cent. It would therefore seem the duty of all who are responsible for steam installation and operation to leave the determination of safety valve size and selection no longer to such statutes as may happen to exist in their territory, but to investigate for themselves.

17 The elements of a better rule for determining safety valve size exist in Napier's formula for the flow of steam, combined with the actual discharge area of the valve as determined by its lift. In Steam Boilers, by Peabody & Miller, these principles in determining the discharge of a safety valve have already been indicated. The uncertainty of the coefficient of flow, that is, of the constant to be used in Napier's formula when applied to the irregular steam discharge passages of safety valves, has probably been largely responsible for the fact that this method of obtaining valve capacities has not been more generally used. To determine what this constant or coefficient of flow is, and how it is affected by variations in valve design and adjustment, an extended series of tests has recently been conducted by the writer at the Stirling Department of the Babcock & Wilcox Co., at Barberton, Ohio.

18 A 373-h.p. class K. No. 20 Stirling boiler, fired with a Stirling chain grate, with a total grate area of 101 sq. ft., was used. This

boiler contained a U-type superheater designed for a superheat of 50 deg. fahr. The water feed to this boiler was measured in calibrated tanks and pumped (steam for the pump being furnished from another boiler) through a pipe line which had been blanked wherever it was impossible to detect and prevent leakage with stop valves and intermediate open drips. The entire steam discharge from the boiler was through the valve being tested, all other steam connections from the boiler being either blanked or closed with stop valves beyond which were placed open drip connections to indicate any leakage. A constant

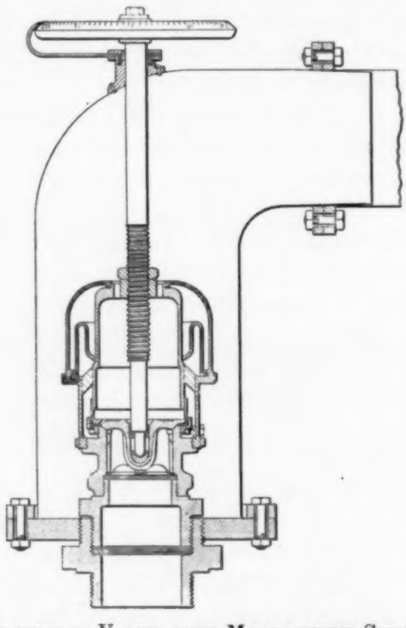

FIG. 6 ARRANGEMENT OF VALVE WITH MECROMETER SPINDLE USED IN THE DIRECT-CAPACITY TESTING AT BARBERTON, OHIO.

watch was kept throughout the testing upon all points of the feed and steam lines, to insure that all water measured in the calibrated tanks was passing through the tested valves without intermediate loss.

19 The valves tested consisted of 3-in., 3½-in. and 4-in. iron stationary valves, and 1½-in., 3-in. and 3½-in. locomotive valves, the latter with and without mufflers. These six valves were all previously tested and adjusted on steam. Without changing the position of the valve disc and ring, the springs of these valves were

then removed and solid spindles, threaded (with a 10-pitch thread) through the valve casing above, inserted. Upon the top end of these spindles, wheels graduated with 100 divisions were placed. Fig. 6 shows the arrangement used with the locomotive valves, the spindle and graduated wheel being similar to that used with the stationary valves. By this means the valve lift to thousandths of an inch was definitely set for each test and the necessity for constant valve lift readings, with that source of error, eliminated.

20 In conducting the tests three hours' duration was selected as the minimum time for satisfactory results. Pressure and temperature readings were taken every three minutes, water readings every half hour. A man stationed at the water glass regulated the feed to the boiler to maintain the same level in the boiler during the test; other men were stationed, one at the water tanks, one firing and one taking the pressure and temperature readings. Pressure readings were taken from two test gages connected about 4 in. below the valve inlet, the gages being calibrated both before and after the series of tests was run and corrections applied. In all 29 tests were run, of which 15 were 3 hours long, 4 were 2½ hours, 3 were 2 hours, and 7 of less duration.

21 Tests numbered 1 to 5 were preliminary runs of but one hour or less duration apiece, and records of them are thus omitted in the following table, which gives the lifts, discharge areas, average pressure and superheat, and the steam discharge in pounds per hour of each of the other tests. The discharge areas have been figured for 45-deg. seats from the formula, $a = 2.22 \times D \times L + 1.11 \times L^2$ where a equals the effective area in square inches, D equals the valve diameter in inches, and L equals the valve lift in inches. In tests 8 and 23, where the width of valve seat was 0.225 in. and 0.185 in. respectively, and the valve was thus slightly above the depth of the valve seat, the area was figured for this condition.

22 As previously stated, the application of these results is in fixing a constant for the flow of Napier's formula as applied to safety valves. The formula is $w = a \dfrac{P}{70}$ in which w equals pounds discharged per second, P equals the absolute steam pressure behind the orifice or under the valve and a equals the effective discharge opening in square inches. This may be stated as $E = C \times a \times P$; in which E equals the pounds steam discharge per hour and C equals a constant. The values of E, a and P being given for the above tests C is directly obtainable.

23 Figuring and plotting the values of this constant indicates the following conclusions:

a Increasing or altering the steam pressure from approximately 50 to 150 lb. per square inch (tests 14 and 10) does not affect the constant, this merely checking the applicability of Napier's formula in that respect.

b Radically changing the shape of the valve disc outside of the seat at the huddling or throttling chamber, so-called, does not affect the constant or discharge. In test 15 the valve had a downward projecting lip (as in Fig. 1), deflecting the steam flow through nearly 90 deg., yet the discharge was practically the same as in tests 10 and 14, where the lip was cut entirely away (as in Fig. 6), giving a comparatively unobstructed flow to the discharging steam.

c Moving the valve adjusting ring through much more than its complete adjustment range does not affect the constant or discharge. (Tests 16 and 17.)

d The addition of the muffler to a locomotive valve does not materially alter the constant or discharge. There is but 2 per cent difference between tests 10 and 13.

e Disregarding the rather unsatisfactory 1½-in. and 3-in. locomotive valve tests, the different sizes of valves tested show a variation in the constant when plotted to given lifts of about 4 per cent.

f There is a slight uniform decrease of the constant when increasing the valve lifts.

24 The variations indicated in the last two conditions are not large enough, however, to impair materially the value of a single constant obtained by averaging the constants of all the 24 tests given. The selection of such a constant is obviously in accord with the other four conditions mentioned. This average constant is 47.5, giving as the formula $E = 47.5 \times a \times P$. Its theoretical value for the standard orifice of Napier's formula is 51.4, of which the above is 92½ per cent.

25 To make this formula more generally serviceable, it should be expressed in terms of the valve diameter and lift, and can be still further simplified in its application by expressing the term E (steam discharged or boiler evaporation per hour) in terms of the boiler heating surface or grate area. For the almost universal 45-deg. seat the effective discharge area is, with a slight approximation, $L \times$ sine $45 \times \pi \times D$, in which L equals the valve lift vertically in inches and

D the valve diameter in inches. Substituting this in the above formula gives $E = 47.5 \times L \times \text{sine } 45 \times \pi \times D \times P$, or $E = 105.5 \times L \times D \times P$.

26 The slight mathematical approximation referred to consists in multiplying the ($L \times$ sine 45) by ($\pi \times D$) instead of by the exact value ($\pi \times D + \frac{1}{2}L$). To find directly the effect of this approximation upon the above constant, the values for E, L, D and P from the tests have been substituted into the above formula and the average constant re-determined, which is 108.1. The average lift of all the tests is 0.111 in. Plotting the constants obtained from the above formula in each test, as ordinates, to valve lifts, as abscissae, obtaining thus the slight inclination referred to in Par. 23 *f*, and plotting a line with this inclination through the above obtained average constant 108.1, taken at the 0.111-in. average lift, gives a line which at a maximum lift of say 0.14 in. gives a constant of just 105. At lower lifts this is slightly larger. Hence 105 would seem to be the conservative figure to adopt, as a constant in this formula for general use, giving

$$E = 105 \times L \times D \times P$$

This transposed for D gives:

$$D = 0.0095 \times \frac{E}{L \times P}$$

Note that the nominal valve area does not enter into the use of this formula and that if a value of 12, for instance, is obtained for D it will call for two 6-in. or three 4-in. valves. For flat seats these constants become 149 and 0.0067 respectively.

27 The fact that these tests were run with some superheat (an average of 37.2 deg. fahr.) while the majority of valves are used with saturated steam, would, if any material difference exists, place the above constants on the safe side. The capacities of the stationary and locomotive valves, the lift test results of which are summarized in Par. 8, have been figured from this formula, taking the valve lifts at opening, and in pounds of steam per hour are as follows:

Of the seven 4-in. iron body stationary valves, the average capacity at 200 lb. pressure is 7370 lb. per hour, the smallest capacity valve (figured for a flat seat) has a capacity of 3960 lb., the largest 12,400 lb., and of the six $3\frac{1}{2}$-in. muffler locomotive valves at 200 lb. pressure,

the average capacity is 6060 lb. per hour, the smallest 4020 lb., the largest 11,050 lb.

28 To make the use of the rule more direct, where the evaporation of the boiler is only indirectly known, it may be expressed in terms of the boiler heating surface or grate area. This modification consists merely in substituting for the term E (pounds of total evaporation) a term H (square feet of total heating surface) multiplied by the pounds of water per square foot of heating surface which the boiler will evaporate. Evidently the value of these modified forms of the formula depends upon the proper selection of average boiler evaporation figures for different types of boilers and also upon the possibility of so grouping these boiler types that average figures can be thus selected. This modified form of the formula is

$$D = C \times \frac{H}{L \times P}$$

in which H equals the total boiler heating surface in square feet and C equals a constant.

29 Values of the constant for different types of boilers and of service have been selected. These constants are susceptible of course to endless discussion among manufacturers, and it is undoubtedly more satisfactory, where any question arises, to use the form containing term E itself. Nevertheless the form containing the term H is more direct in its application and it is believed that the values given below for the constant will prove serviceable. In applying the formula in this form rather than the original one, containing the evaporation term E, it should be remembered that these constants are based upon average proportions and hence should not be used for boilers in which any abnormal proportions or relations between grate area, heating surface, etc., exist.

30 For cylindrical multitubular, vertical and water-tube stationary boilers a constant of 0.068 is suggested. This is based upon an average evaporation of 3½ lb. of water per square foot of heating surface per hour, with an overload capacity of 100 per cent, giving 7 lb. per square foot of heating surface, the figure used in obtaining the above constant.

31 For water tube marine and Scotch marine boilers, the suggested constant is 0.095. This is based upon an overload or maximum evaporation of 10 lb. of water per square foot of heating surface per hour.

32 For locomotives the constant 0.055 was determined experi-

mentally as explained below. Special conditions to be considered in locomotive practice separate it from regular stationary and marine work. In the first place the maximum evaporation of a locomotive is possible only with the maximum draft obtained when the cylinders are exhausting up the stack, at which time the throttle is necessarily open. The throttle, being open, is drawing some of the steam and therefore the safety valves on a locomotive can never receive the full maximum evaporation of the boiler. Just what per cent of this maximum evaporation the valve must be able to relieve under the most severe conditions can only be determined experimentally. Evidently the most severe conditions obtain when an engineman after a long, hard, up-hill haul with a full glass of water and full pressure, reaching the top of the hill, suddenly shuts off his throttle and injectors. The work on the hill has gotten the engine steaming to its maximum and the sudden closing of throttle and injectors forces all the steam through the safety valves. Of course the minute the throttle is closed the steaming quickly falls off and it is at just that moment that the most severe test upon the valves comes.

33 A large number of service tests have been conducted to determine this constant. The size of valves upon a locomotive has been increased or decreased until one valve would just handle the maximum steam generation, and the locomotive heating surface being known the formula was figured back to obtain the constant. Other special conditions were considered, such as the liability in locomotive practice to a not infrequent occurrence of the most severe conditions; the exceptionally severe service which locomotive safety valves receive; and the consequent advisability of providing a substantial excess valve capacity on locomotives in this service.

34 As to the method of applying the proposed safety valve capacity rule in practice, manufacturers could be asked to specify the capacities of their valves, stamping it upon them as the opening and closing pressures are now done. This would necessitate no extra work further than the time required in the stamping, because for valves of the same size and design, giving practically the same lift, this would have to be determined but once, which of itself is but a moment's work with a small portable lift gage which is now manufactured. The specifying of safety valves by a designing engineer could then be as definite a problem as is that of other pieces of apparatus. Whatever views are held, as to the advantages of high or low lifts, there can be no question, it would seem, as to the advantage of knowing what this lift actually is, as would be shown in this

TABLE 1 SAFETY VALVE CAPACITY TESTS

RUN AT THE STIRLING WORKS OF THE BABCOCK & WILCOX CO., BARBERTON, OHIO, NOVEMBER 30 TO DECEMBER 23, 1908

TEST NUMBER	DURATION OF TEST	SIZE AND TYPE OF VALVE	ADJUSTMENT REMARKS	VALVE LIFT	PRESSURE	SUPERHEAT	DISCHARGE PER HOUR	DISCHARGE AREA*	REMARKS
	Hours			Inches	Lbs. per Sq. In.	Deg. F.	Lbs. of Steam	Sq. In.	
6	3	4-in. R. F. Iron Stationary	Regular Adj., Exh. Piped	0.0695	151.7	43.6	5120	0.6226	No Back Pressure
7	3	"	"	0.139	145.4	45.1	8600	1.255	Back Pressure 2 lb.
8	3	"	"	0.180	135.7	49.2	11020	1.704	Back Pressure 3 lb., Max. Lift Test,
9	3	"	"	0.1045	149.4	41.9	7290	0.9400	Back Pressure 1 lb.
10	2½	3½-in. Locomotive Type R	Regular Adj. Without Muffler	0.140	146.7	39.0	8685	1.109	Open Locomotive Valve
11	3	"	"	0.070	152.5	38.0	4670	0.5493	
12	3	"	"	0.105	150.3	41.2	6780	0.8280	
13	3	" Type S	With Muffler	0.1395	146.3	38.1	8400	1.106	Muffler Valve in this and Following Locomotive Tests
14	2	"	"	0.140	52.2	51.3	3620	1.109	Test at Low Steam Pressure
15	2½	Same Except with Lipped Feather	"	0.140	146.4	39.0	8600	1.109	Different Type of Valve Disc
16	3	4-in. R. F. Iron Stationary	Reg. Adj., Exh. Piped	0.140	138.5	42.3	8770	1.265	No Back Pressure, Repetition of Test 7
17	3	"	Adj. Ring One Turn $\frac{1}{16}$-in. above Reg. Position	0.140	142.0	50.1	8900	1.265	Back Pressure 3 lb. Adj. Ring Position Ch'ng'd

18	2	1½-in. Locomotive Type S	Regular Adj., with Muffler	0.107	140.8	23.0	2515	0.4272	Unsatisfactory tests; Valve too Small for Boiler
19	1	"	"	0.060	151.2	None	1550	0.2038	
20	2½	"	"	0.075	146.3	None	2025	0.2560	
21	2½	"	"	0.075	147.7	None	1975	0.2560	
22	1½	3½-in. R. F. Iron Stationary	Regular Adj., Exh. Piped	0.070	146.8	42.6	4320	0.5493	No Back Pressure
23	3	"	"	0.140	139.9	43.6	8360	1.136	No Back Pressure
24	3	"	"	0.105	141.6	48.7	6300	0.8280	No Back Pressure
25	3	3-in. R. F. Iron Stationary	"	0.130	140.1	48.4	6370	0.8846	
26	3	"	"	0.100	142.8	45.6	5160	0.6770	
27	2	"	"	0.070	142.4	29.5	3705	0.4716	
28	3	3-in. Locomotive Type S	Regular Adj., with Muffler	0.130	138.4	48.7	7060	0.8846	
29	3	"	"	0.090	139.3	43.9	4950	0.6084	

* The valves all having 45-deg. bevel seats these areas are obtained from formula

$$a = 2.22 \times D \times l + 1.11 \times l^2$$

except as in tests 8, 23, where the valve lift is greater than the depth of the valve seat, in which case the following formula is used:

$$a = 2.22 \times D \times d + 1.11 \times d^2 + \pi \times D \times (l - d)$$

a = Discharge area. l = Valve lift.
D = Valve diameter. d = Depth of valve seat.

specifying by manufacturers of the capacity of their valves. Further, as to the feasibility of adopting such a rule (which incorporates the valve lift) in statutes governing valve sizes—this would involve the granting and obtaining by manufacturers of a legal rating for their valve designs based upon their demonstrated lifts.

35 This paper has dealt with but one phase of the subject of safety valves in order that its conclusions might be drawn more clearly. The apparatus and tests shown indicate that the lifts and capacities of different make valves of the same size and for the same conditions vary as much as 300 per cent, and that there is therefore the liability of large error in specifying valves in accordance with existing rules and statutes, because these rules as shown rate all valves of one size as of the same capacity, irrespective of this variation. A simple rule is given, based upon an extended series of direct capacity tests, which avoids this error by incorporating a term for the valve lift. Finally, the method and advantage of applying this rule in practice has been briefly indicated.

DISCUSSION ON SAFETY VALVES

MR. LUTHER D. LOVEKIN.[1] The subject of Safety Valves, while well worth the attention of the profession, has received little or no consideration from the average engineer. Various rules have been employed for determining the size of safety valves, some of which appear too crude to have been even considered, much less adopted. Of late, however, considerable attention has been given to this subject; and to my knowledge numerous tests have been made that should prove extremely interesting.

2 We all know that the function of the safety valve is to prevent the pressure of steam from rising to a dangerous point, and in order to accomplish this, the effective opening of any valve should be sufficient to discharge all the steam that the boiler can generate.

3 During the year 1903, I was asked to look into the rules and regulations concerning safety valves prescribed by the United States Board of Steamboat Inspectors. The rule in use was based on grate surface, without regard to the amount of coal burned in a given time. It served its purpose without trouble since such a thing as forced draft was then almost unknown. At the present time, however, when we are confronted with the problem of building boilers for all purposes, some of them required to burn 12 lb. of coal and others as high as 70 lb. per square foot of grate, the question of the proper size of safety valve can be determined only on the amount of evaporation.

4 Having this in view, I prepared a new set of rules based on Napier's well-known formula for the flow of steam through an orifice; concluding that while the opening through a safety valve cannot be considered as having exactly the same effect as an orifice, the difference would be so slight as to be negligible. These were presented to the Board of Supervising Inspectors at their annual meeting in Washington, and adopted by them as the standard, after careful examination. The derivation of the formula is given below:

Let

P = absolute pressure.

W = weight discharged per hour in pounds.

[1] Mr. L. D. Lovekin, Chief Engineer, N. Y. Shipbuilding Co., Camden, N. J.

A = area valve opening in square inches.
d = diameter of valve in inches.
a = area of valve of diameter d.

From Napier's rule

$$W = \frac{360\ A\ P}{7}$$

For safety-valve practice allow 75 per cent of this and restrict the lift of valve to $\frac{1}{32}$ diameter.

Then

$$W = \frac{270\ P}{7} \times \frac{\pi d^2}{32} = 4.821\ Pa$$

$$a = 0.2074 \frac{W}{P}$$

If W represents weight of water in pounds evaporated per square foot of grate surface per hour the above formula will give the area of valve required per square foot of grate surface.

5 It will be noted that in preparing this work, the lift was based on $\frac{1}{32}$ of the diameter of the valve; and while I considered this lift within good practical limits, I have found that a number of different safety valve manufacturers differ with me in this regard. Whether the valve is restricted to $\frac{1}{32}$ of its diameter or not, however, the net area of the opening should in my mind be at least equal to that shown in the table.

6 I am not in favor of what might be termed "an excessive lift of valve," such as one-fourth of the diameter, although some of the best authorities in the inspection of steamships still ask for that lift, the British Board of Trade being one of the foremost who do so. It should be said in favor of the rules of the British Board of Trade, as well as of Lloyds and the Bureau Veritas that they demand an accumulation test in connection with all spring-loaded safety valves; as follows (Page 80, par. 128, "Spring Loaded Safety Valves to be Tested under Steam"):

7 "In no case is the surveyor to give a declaration for spring-loaded valves unless he has examined them and is acquainted with the details of their construction, and unless he has tried them under full steam and full firing for at least twenty minutes, with the feed water shut off and the stop valve closed, and is fully satisfied with the result of the test. If, however, the accumulation of pressure

exceed ten per cent of the loaded pressure, he should withhold his declaration and report the case to the Board."

8 This is undoubtedly one of the safest methods to follow in determining the size of a safety valve, as the results will be based upon facts and not upon figures.

9 There is no doubt that at the present day it is crude in the extreme to fix the size of a safety valve by specifying the diameter simply, regardless of its lift and consequently its relieving capacity; and the practice should be corrected. I would not wish to restrict the makers of valves to any one particular lift, unless, of course they could come to an agreement as to the advantage of certain definite lifts.

10 Unfortunately in accepting my formula and table of safety valves, the board failed to state in their rules that the sizes were based upon a lift of valve equal to $\frac{1}{32}$ of its diameter; and consequently have left out a most important element which I shall ask in the near future to have inserted. Following the rules of the Steamboat Inspection Service as they now exist in printed form, it is quite possible for a valve to be of the proper size in inches, and yet fall far below the actual requirements.

11 Having settled upon the proper diameter for a safety valve, according to the above formula, it will be evident that the clear area between the valve and its seat (due to having a lift equal to $\frac{1}{32}$ of its diameter) is only about $\frac{1}{11}$ of the area of the nominal diameter found by said formula. Therefore it would seem that the inlet from the boiler to the safety valve should be equal in area only to the free area between the safety valve and its seat; this would reduce the opening in the boiler to about $\frac{1}{11}$ of the area used at the present time. Experiments in this line, however, have shown that a free entrance from the boiler to the safety valve is absolutely necessary to prevent chattering. Just what this relation is I have not determined; in fact, it would depend entirely on the length of the nozzle or pipe connecting the safety valve to the boiler. In most cases safety valves are bolted either directly to the boiler or to a casting which is bolted directly to the boiler, and which forms a seat for both the safety and the stop valves, so that there would be very little to gain in reducing the inlet nozzle to a safety valve.

12 While dealing with the inlet side of a safety valve I wish to bring out a feature seldom if ever discussed in connection with safety valves, and that is the question of placing the safety valve upon the outlet end of dry pipes in boilers. These dry pipes, as is well known,

usually run along the upper part of a boiler and have slots cut in to give an area equal to the full area of the pipe. In some cases I have found the steam pressure within the boiler to be 200 lb. per square inch, while that at the outlet of the dry pipe was only 180 lb. per square inch; in other words, a drop of 20 lb. pressure took place, due to wire-drawing the steam through the slots in the dry pipe.

13 In other cases we have had 300 lb. boiler pressure in connection with water-tube boilers, and have purposely restricted the flow of steam through the dry pipe so as to cause a reduction in pressure of 50 lb. per square inch, and thus obtain a slight degree of superheat. In this latter case, however, the safety valves were applied to the boiler drum, and not in connection with the dry pipe.

14 This question of fitting dry pipes, and connecting the safety valves to their outlets, is one which should receive careful consideration, and I have found it advisable, in order to prevent excessive pressures in the boilers, to have at least 25 per cent excess area through the slots.

15 Assuming for the present the inlet of the safety valve to be of approximately the same diameter as the valve itself; so as to prevent chattering, the point of the next greatest importance is the outlet area from the safety valves; some rules insist on an outlet area equivalent to the full bore of the safety valve, which appears both inconsistent and unreasonable; for if we have only $\frac{1}{11}$ of the area for the steam to pass through at the valve seat, we certainly do not require the full area for the steam to pass to the atmosphere.[1]

16 It is almost impossible to make any close calculations for this that would suit all conditions, as a study of the various arrangements of safety valve escape pipes will show; but I think that an outlet from a safety valve equal to $\frac{1}{2}$ the area of the valve would no doubt suffice in all cases. In fact most of our United States battleships are equipped in this manner, and I think this should receive the attention of the board and others.

17 While the U. S. Cruiser Tennessee was on trial the main engines were stopped suddenly, due to trouble with one of the connecting rods. All the safety valves responded instantly, and though the steam pressure went up to 10 lb. above popping point, no trouble was experienced as the result of this sudden stoppage; proving beyond doubt that a combined area of outlet pipes equal to $\frac{1}{2}$ the area of the safety valves was sufficient to prevent an excessive

[1] The area of safety valves referred to above is the nominal area of the valve.

accumulation of pressure. This seems reasonable and might have been true with less area; a matter, however, that can be determined only by actual experiment.

MR. ALBERT C. ASHTON. It seems to the writer that what is most needed today is not necessarily that standard makes of pop safety valves should have a greater capacity of relief than is now furnished by them, but rather that there should be a better understanding of the proper proportioning of safety valves to boilers, for which no universal rule has as yet been recognized and adopted.

2 Mr. Whyte lays some stress on recent tests that have been made to determine the comparative capacity of pop safety valves now on the market, being doubtless, the tests referred to by Mr. Darling. While these tests show what such valves will accomplish under certain favorable conditions they have not clearly demonstrated that high-lift valves so made are a success on all applications. They certainly have shown many failures in locomotive service during the past year, and must still be classed as an experiment.

3 Under some circumstances they may be considered even dangerous, as in cases where they have lifted the water from the boiler, caused leaky tubes, as well as where they have gone rapidly to pieces on account of their harsh action.

4 Safety valves should never give such a large and sudden relief as to affect the water level in a boiler, neither should they close so suddenly as to be a shock to the boiler by the quick stoppage of the flow of steam. High-lift valves which do this are not as practical as lower lift valves which give a somewhat slower and easier relief. In other words, the most satisfactory safety-valve service is that which causes the least shock to a boiler, and yet controls closely any possibility of over-pressure. These factors should be considered in the discussion.

5 If high-lift valves were certainly an improvement over the best standard makes, safety valve manufacturers in general would change their designs, as can easily be done, and make nothing but high-lift valves The tests which Mr. Darling has explained tonight show an average lift of $\frac{1}{8}$ in. for the high-lift valves which is about double the lift of standard valves. Such high lift seems to the writer excessive, although there may be some virtue in making valves with a little higher lift than the common standard of $\frac{1}{16}$ in.

6 It is to be hoped that as a result of this meeting the Society will take up for investigation the broad question, as to the most practical

schedule or formula for determining the proper capacity of relief that safety valves should give on various-sized boilers at various pressures.

Mr. A. B. Carhart. The spring is the heart of the modern pop safety valve. The operations of a poorly-designed valve may be greatly assisted by a suitable spring, and a valve with an excellent arrangement of the other parts may be seriously handicapped by an improperly designed spring, or even transformed from a safety device to a source of danger.

2 In making a safety valve spring there are practical limitations which must be taken into account. Steel of very large diameter cannot be wound satisfactorily upon a small mandrel. A spring excessively long in proportion to its diameter and pitch may bend or buckle instead of compressing in a straight line axially; and if the number of coils be too great the reaction of the spring will set up an oscillation which will cause destructive chattering of the valve. The valve disc must not, in effect, be suspended at the end of a flexible spring, but must have behind it at all times a positive force capable of controlling its action when lifted by the escaping steam. If the spring be too short, not only will the reaction be too sudden, but the active free coils will form so small a proportion of the total length that the spring compression will be greater on one side than on the other and there will be an undesirable sidethrust on the disc guides. If the pitch be steep or the coils wound far apart, there will be room for considerable free movement and apparently a sufficient deflection of each coil to provide for the desired compression; but the fiber stress will be enormously increased and the rod may be fractured or a permanent set produced in it. On the other hand, if there are too many coils, there will not be sufficient free space between them to permit even the small compression necessary, and the spring will have insufficient reactive power.

3 The performance required of the spring in a pop safety valve is different from that expected of car springs or similar buffers. In carrying the load of a car or wagon, any unevenness of the road causes a jolting or bouncing, and the momentum of the moving car adds temporarily to the effect of its dead weight in increasing the violent action of the spring. Under severe conditions such springs are often compressed to their limit, until the coils are in solid contact, and a severe bump or jar is felt in the car itself; but the reaction of the spring, when the unevenness is past, sets the car back to its proper

position, and on the rebound it may be that the car has risen above its normal position so that the spring is drawn out in tension; after a while these waves of oscillation subside, and the car rides normally. Car springs are generally designed so that when the car is loaded in an ordinary manner the spring will be about one-half compressed. This gives some leeway for additional load in the car, and also permits the greatest movement possible above and below the normal medium position. It is believed that the condition of longest life is met when the calculated ordinary load will compress the spring one-half of its total free movement.

4 For pop valve service, on the other hand, the exact pressure load is determined by the exposed area of the disc and the steam pressure; and when the valve opens there is an additional load governed by the additional exposed area of the disc and the steam pressure, which rapidly decreases as the steam directly below the valve escapes and the boiler is relieved. Under no conceivable conditions of actual service can sufficient steam pressure be brought upon the disc to compress the spring so that the coils will be solid, metal to metal, if it has been well designed for its original fixed load; and the additional spring compression to permit the valve's opening in order to relieve the boiler is comparatively little; possibly 0.08 in., more commonly and preferably less, and never under any conditions as much as 0.12 in., or say $\frac{1}{8}$ in. the extreme lift. If after the fixed load-pressure is reached the spring has still $\frac{15}{32}$ of unused possible compression, of which less than $\frac{3}{32}$ in. will be required to accommodate the desired lift of the valve, there will still be $\frac{3}{8}$ in. more before the spring will go solid, at which point all further compression would be impossible; therefore the valve spring can be properly designed to carry its set load at much more than half of its total free compression, and nearer to its solid condition than would be wise with a car spring where the amount of motion is not limited.

5 If springs are properly proportioned, the point of greatest resilience, elasticity and reaction, securing sharp action in the valve and accurate adjustment of opening pressure, is in the last one-third of the total possible free compression, and this is the part of the spring action which should be utilized for safety valve service. I believe it proper to proportion the spring so that the set load is carried at about two-thirds or three-fourths of its total free compression, making the dimensions such that the remaining unused compression of the spring is ample for the lift of the disc and a safe margin beyond.

6 While it may be said that the springs will never be subjected to the extreme compression required to force them solid in service, yet where the working compression is such a large proportion of the total free movement, the spring might be dangerously near the point of setting or fracture unless proportioned and tempered to take the solid test. In making boiler tests the head bolt may be set down until the spring is solid, to close the valve; and if the valve is fitted with a lever, the spring may at any time be lifted or compressed an indefinite amount by that means, even to solid. I would not consider it proper to use in a valve with a lever any spring that would not safely take the solid test and was not capable of being compressed until the coils are metal-to-metal, any number of times consecutively, without showing any permanent set.

7 Otherwise there can be no mark or means to prevent the spring from being screwed down, even through ignorance, until the danger point may be once passed, and the spring then takes a permanent set, after which it becomes entirely ineffective as a valve spring, and a source of danger if its use is ignorantly continued.

8 One prominent manufacturer of safety valves requires all springs to be designed to take this solid test indefinitely. After the springs are made and tempered they are closed solid in a press at least three times; and again before they are put into valves for service, at the time when the ends are dressed and fitted, they are again tested three times in the same way and the compression is noted at the load for which they are designed. If any spring sets, even temporarily, as much as $\frac{1}{16}$ in., or if there is much variation in the compression at normal load, it is condemned and rejected. Out of the great number of valve springs made and tested every year under these conditions less than one-third of one per cent are rejected, which shows the requirements to be commercially practicable and the method of calculation and design sound and conservative.

9 Springs of comparatively poor design, if well made of proper steel, heated uniformly to the temperature for working and tempered skillfully, are better than springs of better dimensions but improperly made. In most small shops where springs are wound by hand, the long bar of steel must be heated in short sections in a small furnace or forge fire, winding about a foot of the steel at each operation, drawing the bar by hand-tongs around a cold mandrel and then sticking the remainder of the straight bar again into the fire until it is soft enough to wind another coil or two. Each foot of the steel has then been heated to a different temperature, and where the heated sections

have overlapped, the steel is likely to be burned. The same difficulty arises in tempering, where it is necessary to secure even heating Bars measure from 5 to 12 ft. long for valve springs of the largei sizes, and for locomotive valves the lengths would be about 4 to 6 ft. The whole bar should be evenly heated at one time, without being exposed to any direct de-carbonizing flame or forge blast, then wound accurately and quickly at one operation, and plunged in the tempering bath at exactly the proper moment.

10 These springs should be wound of a special grade of steel, kept up to standard specifications, in the various sizes and shapes of bar required for different loads and pressures; and although they are wound in the same general manner and have much the same outward appearance as ordinary coiled helical springs used for car springs and other commercial purposes, in reality they are very different in treatment and character, and proper results can never be obtained if springs of ordinary steel are substituted. No railroad should attempt to use car springs of similar shape that may be on hand, or to buy springs for safety valves by specifying simply the measured dimensions.

11 In measuring springs, it is the custom and better practice to state first the inside or mandrel diameter, then the free length of the finished spring and last the diameter and form of the steel rod used, the measurement of the cross-section being that of the straight bar before winding. For long springs it is sometimes necessary to use a taper mandrel to facilitate the drawing-off of the spring, although generally the slight natural expansion of the spring after winding will sufficiently release it; where the mandrel is tapered, the mean diameter, approximately half-way between the two ends, is the dimension to be specified. The over-all outside diameter will generally be slightly more, and the outer face of the coiled square steel will be less, than the commercial dimensions would indicate.

12 It is plain that we can get no more work out of a spring in reaction than is put into it, and the more compression put into a spring, the more work it will return, and the sharper and more positive will its action be in controlling the valve. The way to get proper work out of a spring is to put force into it in effective stress.

13 Low fiber stress is not a measure of safety but of ineffectiveness. To develop properly the resilience of the most trustworthy and suitable steel available today requires a stress that would be inadmissible with material of inferior temper or uncertain quality, and subject to ordinary commercial defects. Experience shows that springs may best be stressed at from 60,000 to 75,000 lb. per sq. in.

at the fixed load, which should compress the spring about 70 per cent of its total possible free movement. The remaining movement should be three or four times the further lift of the valve in opening. This is a low stress for the material and gives safe working limits. Under the same formula the stress when the spring is solid will not exceed 100,000 lb. per sq. in.

14 The limit of elasticity, beyond which a permanent set occurs, is different for torsion and for elongation and independent of the tensile strength, and for steel of the proper characteristics for spring-making this torsional elasticity is relatively high. Car-springs have sometimes been calculated upon a fiber stress as low as 30,000 lb. per sq. in. at normal load, and this may have seemed reasonable for common grades of steel under circumstances where the springs might frequently be subjected to extreme compression or extension. But with the material used today a fiber stress of 80,000 lb. is generally recommended. This is not a question of keeping within limits of safety, but of stressing the steel to its point of proper efficiency. It would be absurd to expect a spring suitable for use in a valve at 200 lb. pressure to show proper performance and lift, or permit satisfactory opening and closing of the valve, at only 50 lb. pressure; its reaction and resilience could not be at all reasonably developed.

15 If a spring is designed for a fiber stress of 70,000 lb. per sq. in. at normal load, and as much as 100,000 lb. per sq. in. when compressed solid, and is made of such steel that it can remain assembled in the valve indefinitely under pressure without perceptible set, and can be compressed solid an indefinite number of times without injury, it is evident that it is used at a safer percentage of its elastic limit than springs made of less virile steel, which although designed for only 30,000 lb. per sq. in. fiber stress at normal load, will suffer a gradual set or deterioration in service or will become permanently set if tested solid.

16 Springs of bronze are notoriously inefficient and unenduring, and their depreciation and permanent set in service at comparatively low fiber stress more than balance any possible advantage of slow corrosion; and certain grades of steel may be as poorly adapted for the making of valve springs. The torsional elasticity and power depend not upon the tensile strength so much as upon the temper and resilience. Therefore some of the new alloy steels have proved disappointing for this service and the name of any alloy can not as yet be used either as a fetish or a selling-phrase.

17 Observation of many thousand springs in continuous daily

service, under severest conditions of constant use, shows that springs calculated upon a very high fiber stress are entirely reliable for indefinite periods of service and do not develop any measurable percentage of faults or fractures; the failure of valves under operating conditions attributable to such springs is practically unknown and is less than is due to the rare defects of the other structural parts.

18 Large movement of the spring in compression is undesirable; it is but a necessary means to an end, an evil to be kept within minimum limits. It would be an advantage if a satisfactory discharge area of the valve could be attained with even less spring compression than at present. Large lift of the disc is not a measure of capacity but of inefficiency; for the valve which releases the steam with the least proportional lift or spring compression is to that degree the more efficient for its purpose and at the same time more safe and reliable.

19 The specifications which require valve seats to be made of non-corrosive metal, and the rules which compel every valve to be tried and lifted by the lever every day, aim to overcome the ever-present danger that the valve may stick upon its seat and fail to open at the critical moment; but the greatest cause of the sticking of the valve, when it does occur, is not corrosion of the seat face but the binding friction of the disc-guides against the side of the well or throat of the valve. This cocking or binding effect can be decreased by any modification of design which will reduce the diameter of the cylindrical guide, or which will bring the guiding surface close to the plane of the seat, both of which would reduce the moment of the friction or cocking stress. Any device which reduces the lift of the disc and the spring movement to the least possible amount will also reduce the eccentric spring action and its effect, and of course any valve design which contemplates an unnecessarily large lift or compression disadvantageously magnifies this effect. It is perhaps important to point out that as the primitive and still common form of safety valve has a seat opening beveled at an angle of 45 deg., the effective passage through the seat is measured by the sine of 45 deg. and is approximately only 0.7 of the actual vertical lift or compression of the spring when the valve opens, so that the spring must necessarily compress about $1\frac{1}{2}$ times the effective lift, and even this does not always afford a free passage for the steam to the air where there is vertical overlap of the regulating ring against the lip of the disc in order to increase the lift against the greater pressure of the shortening spring.

20 In the well-known annular type of valve, the area of the disc

open to the constant pressure of the steam is approximately only four-fifths of the total initial area of the disc under load in the bevel-seated form of valve having the same diameter and seat circumference, therefore the use of the familiar annular, flat-seated valve is the logical way to reduce to a minimum all the difficulties of spring making, especially where the space available for the spring is absolutely limited by the over-all dimensions permitted by locomotive builders and boiler-makers; for the spring need thus be of dimensions and strength to carry only four-fifths of the load necessary in the lip-type of valve; the vertical lift and spring compression require to be only 0.7 as much, or for the same lift will give 1½ times as much discharge area. No preliminary lift is required to relieve the overlap of an adjusting ring, for the work of giving to the disc its sudden pop lift is performed by an auxiliary steam discharge by-passed through the central passages. This by-passed or auxiliary discharge adds its volume to the main discharge capacity and leaves an absolutely unrestricted and unthrottled free escape for the main flow of the released steam directly to the open air, without any tortuous expansion chamber or deflecting ring; the outlet is across a flat seat which not only utilizes the full vertical lift but gives a discharge opening of cylindrical form with efficient rounded edges, and has the further advantage that it is impossible for it to jam or stick; it is easily refaced by rubbing on a face-plate instead of grinding to a bevel, and as the disc can be made entirely without the ever-cocking and sticking guides, the spring can be utilized to the greatest possible degree. My emphasis upon this point of spring limitation and the helpful effect of suitable valve-seat design is because attention has not been generally called to its importance and experience has shown the serious difficulties which may be thus minimized if not entirely avoided.

21 It is not within my purpose to recommend any definite formula for the calculation of helical springs. My investigations have led me to believe that there are not in this country today many men who have experienced the fortuitous concurrence of time, inclination, business necessity and proper manufacturing and testing facilities to lead them to develop a practical working formula very far beyond the very unsatisfactory rules laid down in the handbooks. Experiments have been carried out upon springs wound of comparatively small wire, but everyone who has had occasion to use the conventional formulae must have realized that no matter how well they cover a few types of car-springs within a limited range, they lead us far astray in this special branch of the problem; especially in cases where

the limitations upon the free length of the spring will not permit the use of either round or square bars and some special flat or rectangular section must be used to secure the required area of steel and still leave room for movement between the coils. One engineer to whom I applied for advice replied:

22 "I would say, in reference to the question raised by you, that we do not consider it good practice to allow a stress of over 95,000 or 100,000 lb. for valve springs, although we have in our various experiences stressed springs as high as 145,000 and 150,000 lb., but we do not under any circumstances recommend this or any approximate stresses for the extreme service to which valves are subjected. A long time ago we demonstrated that while various published formulae are correct within certain limits for springs made of round steel, they were far from correct for springs made of squares and special sections."

23 Nearly every engineer of education and experience would be qualified to suggest at least one apparently simple and more or less obvious improvement in the design of safety valves or springs, but nearly every such possible detail will be found to be old. Almost every conceivable device or modification has been the subject of a patent, and most of these have been thoroughly tried, with much expense and enthusiasm, before being condemned and discarded. The various subterfuges of double or concentric springs, one more flexible than the other, of spiral springs with coils of increasing diameter, whose first movement in compression is rapid until the smaller and stiffer coils are brought into action, springs suspended in all sorts of universal bearings, and every method of end bearing and fitting, have all been tried and abandoned by almost every maker.

Mr. E. A. May.[1] In view of recent legislation by various states and municipalities with reference to boiler inspection, and the size of safety valves, this question is assuming considerable importance in the minds of manufacturers of low-pressure cast-iron house-heating boilers. Mr. William Barnet Le Van's valuable treatise on Safety Valves, their History, Antecedents, Inventions and Calculations, has made it seem unnecessary to consider the various formulae for calculating the size of safety valves, and it will rather be the purpose of the writer to bring into prominence the various factors which enter into the make-up of the algebraic equations and which determine the proper size of safety valves.

[1] Mr. E. A. May, Mechanical Engineer, the American Radiator Co., Chicago.

2 There are so few data available applying strictly to low-pressure heating boilers that original research is necessary and it is unfortunate that when such investigation is undertaken by any boiler manufacturer, the data secured are often considered prejudiced by manufacturers of other types of boilers.

FUNCTIONS OF THE SAFETY VALVE

3 Many diverse opinions exist as to the functions of the safety valve, and the question should be discussed from as many viewpoints as possible that there may remain at least a groundwork upon which to build a theory which shall be logical and fully demonstrable.

4 Whether a safety valve should be called upon to exhaust all the steam generated by the boiler at its maximum capacity, or only to care for the excess generated over and above a predetermined amount, offers opportunity for profitable discussion, and this point must be established in order to form a standard for general practice by manufacturers of low-pressure boilers. Practice has demonstrated that a safety valve on a low-presssre boiler is rarely called upon to exhaust all the steam-generating capacity. This is due to several conditions:

- *a* In the majority of heating plants, the full amount of radiation is almost always in service, caring for a large percentage of the steam generated, and even when the radiation is nearly all cut out there is still circulation through the piping.
- *b* Practically every steam boiler used in low-pressure work, which rarely calls for gage pressure in excess of 2 lb., has its damper regulator which, when properly set, checks combustion when 2-lb. pressure is reached.
- *c* Chimney conditions in the majority of heating plants make it almost impossible to drive the boiler to its maximum steam-generating capacity, i.e., the maximum capacity obtainable with every condition exactly right.

5 In practically all house installations at least two of these conditions exist, and in a majority all three, so that we would have to select a valve out of all proportion to actual requirements in order to exhaust all the steam which might be generated by the boiler under its full steam-generating capacity and under ideal conditions.

MAXIMUM CAPACITY

6 It may be argued that if one boiler is installed under ideal conditions, a valve suitable for those conditions should be installed on all boilers of the same size. This brings us to a consideration of maximum capacity and how it is established: whether (1) by the heating surface of the boiler alone; (2) by the grate surface; (3) by the fuel-carrying capacity; (4) by the rate of combustion; or (5) by all combined. Scarcely any two manufacturers of low-pressure house-heating boilers agree in this particular. One may rate solely on the area of heating surface, another on the grate surface, and still another on the amount of fuel the grate will carry, but the writer believes that none of these factors should be considered alone.

7 Many of the rules in force as to the proper size of safety valves are based on the amount of grate surface contained in the boiler. The office of a grate is to furnish a support for the fuel and to admit air for combustion. The fuel-burning capacity of 1 sq. ft. of grate, where the chimney flue is of ample capacity, is controlled by the size and length of the internal gas passages or flues and the quantity and disposition of the heating or absorbing surface. Where the grate surface is too large, air in excess of requirements is likely to enter the fuel and cool the gases liberated by combustion. If there is excess flue surface, giving long gas travel with the attendant frictional resistance, insufficient air enters the fuel chamber, and the latent heat or stored energy of the fuel is not fully liberated.

8 Large grate area and indifferent draft form a bad combination, making it impossible to maintain good combustion over the entire area of grate. In such cases, the size of grate should be reduced, concentrating the draft on a smaller area. It is the writer's opinion that grate area alone is not the basis on which to establish the size of safety valves.

9 While the amount of steam generated is in direct relation to the rate of combustion (i.e., the amount of coal burned per square foot of grate per hour), yet if different types of boilers were selected of practically the same grate area, the writer is confident no two would have the same capacity. In one the amount, position, and arrangement of the heating surface might make it impossible to burn more than 7 lb. of fuel per square foot of grate per hour, with 6 lb. of water evaporated per pound of fuel, which is below normal. Another might have a maximum rate of combustion of 10 lb., with an evaporative power of 8 lb. of water, and another a maximum rate

of combustion of 12 lb., with an evaporative power of 9 lb. of water. It would be unreasonable either to expect or require all three boilers to have the same size safety valve. Before a definite rule as to the size of a safety valve is formulated, therefore, the factors to be used in establishing the maximum rate of combustion and calorific value should be determined.

NEED FOR A COMMON BASIS

10 It is not the purpose of the writer to go into the merits or demerits of boiler rating as especially applied to cast-iron house-heating boilers; but it would seem that there should be a common basis arrived at by manufacturers of house-heating boilers for their ratings, before any attempt is made to establish the size of safety valve required. The relation of valves for low-pressure heating work to the actual capacity of the boiler is so close than one can scarcely be considered without the other.

PROPER SIZE OF THE SAFETY VALVE

11 In view of the wide variation in methods employed by manufacturers in ratings of boilers, as well as in the rules employed by users of safety valves, it would be difficult to select a proper size valve based on grate dimensions only. If valve manufacturers would indicate, in addition to the size of the valve, its capacity at different adjustments for exhausting steam, it would help materially, not only from the standpoint of the boiler manufacturer but of those whose duty it is to inspect the safety valve, and would further aid in the matter of legislation. Valves could in fact be designed and sold on their exhaust capacity without regard to specific size, i. e., owing to variation in design, one valve might have a larger diameter with a lesser lift than another, while their capacity for exhaust might be identical.

12 The simplicity of this method will be appreciated by anyone considering the rules and formulae in effect at present. If the law specified, however, that for a certain evaporative power or rating of boiler a certain exhaust capacity should be maintained in the valve, each manufacturer could determine for himself the proper valve to use.

Mr. H. O. Pond. The engineer about to design a boiler installation finds himself confronted by an array of rules covering the appli cation of safety valves, no two of which, if applied to a case, will give the same results, and the correctness of any of which may be questioned. The size and number of safety valves installed with any boiler have depended in the past, therefore, either upon the person making the installation or upon a more or less effective police or insurance regulation.

2 This practice has not perhaps been as serious a menace to life and property and the proper operation of plants, however, as it has recently become. For a number of years the tendency has been to force boilers further and further beyond the standard ratings, and to get the maximum possible capacity out of the boiler installation, so that valves which may have been of the right size for boilers operating on low ratings undoubtedly would not be correctly proportioned for boilers forced to capacities as high as 200 per cent of rating. The use of the superheater has also introduced an additional factor which must be considered when deciding upon a safety valve installation.

3 The absolute absence of reliable data relative to safety valve performance, and the proportioning of valves for a given service, was brought forcibly to my attention something more than a year ago, in connection with the design of some special boilers of large capacity, equipped with superheaters. When asked for data relative to the capacities of their valves, none of the manufacturers were able to furnish any definite information.

4 It is necessary to know how much steam can be discharged through a given valve at a given pressure and temperature, in order to properly determine the size or number of valves to be used under given conditions. The quantity of steam which can be discharged through valves of the same "catalogue" size, but of different design and manufacture, undoubtedly varies considerably. No two manufacturers use just the same lift of valves of the same catalogue size, nor is the design of seat, muffled ring and ports the same. These points must necessarily effect the discharge through the valve, and they are not properly considered in the present rules governing safety valve practice.

5 What we must know in order to apply safety valves to boilers intelligently is, above all, how many pounds of steam at a given pressure and temperature can be discharged in a given time through the particular valve which it is desired to use; some idea can then be gained as to the number of valves to be installed.

6 Feeling the necessity of getting reliable data on this subject, and that if possible the rules governing the use and application of safety valves to boilers should be revised and brought into rational working form, I have taken the matter up with several of the valve manufacturers and with the Babcock & Wilcox Co., to see if some reliable quantitative tests could be made, and the results used in checking the present rules or formulating new ones for determining the proper proportioning of safety valves to meet boiler conditions. From the papers it will be seen that considerable work has already been done with a view to obtaining some definite information relative to safety valve capacities, and other tests and experiments along the same line are now being conducted. It is hoped that through the Society a movement may be started to revise the rules governing the use of safety valves on steam boilers, and that it may also soon be possible to obtain from the manufacturers a statement of ratings of valves based on capacity of discharge rather than on catalogue size.

Mr. F. J. Cole. Several reasons may be cited in explanation of the apparent disregard of definite rules governing the application of safety valves to locomotives. As this condition exists not only in this country but in foreign lands, it may be of interest to quote from the letter of a prominent locomotive builder abroad:

> I may say that we do not, in fact I do not think any locomotive builders in this country, either private firms or railways, adopt any special practice with regard to size and capacity of safety valves for locomotive boilers.
>
> You are probably aware that the "Ramsbottom" duplex safety valve is almost universally adopted in this country and, with the exception of your own, almost throughout the world. When these safety valves were introduced on the London & North Western Railway in 1858, they were made 3 in. in diameter at the seat, each. In order to show you how little regard is taken in this country to any proportion of area of safety valves to any other part of the locomotive, the size of safety valve I name has been perpetuated, notwithstanding the fact that boilers have nearly double the capacity, and the pressures have been increased 50 per cent. Further, as an illustration of the perfunctory manner in which this matter of safety valves is being dealt with, we have constructed boilers of capacity of no more than that on which two 3-in. safety valves have been fixed with two sets of duplex safety valves of 4½ in. diameter, which means, of course, four separate valves, each that diameter. As a matter of fact, we do not attach much importance to the capacity of a safety valve, as we are very particular with our drivers in censuring them for allowing their engines to blow off steam to any extent. We think a safety valve should be looked upon as merely an instrument to indicate that more water should be injected into the boiler, or less fuel should be put on, and in practice, careful drivers seldom allow their engines to blow off

steam. Exception is made in cases where locomotives may be working hard with a heavy load up-hill and with the necessary fierce fire, coming unexpectedly upon signals against the driver, in which case the shutting of the regulator necessarily involves an accumulation of steam which the safety valves have to carry away.

2 It is probable that general foreign practice for locomotive safety valves is systematized no more than in England or America. While definite rules to govern this matter are desirable, it is evident that on account of the peculiar conditions governing the draft on locomotives, the same necessity does not exist for safety valve regulation as in the case of marine or stationary boilers, the action of the exhaust automatically taking care, in a large measure, of the generation of steam.

3 Usually in urging a locomotive up a steep grade, the boiler is taxed to its utmost capacity, and the water is frequently lowered so as just to show in the lower gage cocks when the summit is reached. When steam is shut off the water is still further lowered, and it is usually necessary to put on the injectors and fill up the boiler, and this same practice has often to be resorted to when an engine is shut off suddenly, on account of a signal unexpectedly displayed, or from other causes. Furthermore, locomotive boilers are so carefully constructed, with a large factor of safety, ranging from 4 to 5, that they have ample margin of strength, and there is no cause for alarm even if the pressure goes temporarily 20 or 25 lb. above the normal blowing-off point.

4 With steam at 200 lb. pressure, the temperature is about 387 deg. fahr. At 220 lb. the temperature is about 395.5 deg. or 8.5 deg. higher. If the pressure temporarily goes 20 lb. above the normal, it means that the entire mass of water has been heated 8.5 deg. higher than before, and it is altogether probable that with this increase in pressure and temperature, most of the radiant heat in the firebox has been absorbed.

5 The writer is in favor of a thorough investigation, looking towards the formulating of definite and authoritative rules for the application of safety valves to locomotives, and he invites attention to the following suggestions for their preparation:

a The diameter, number and kind of safety valves to be based on their capacity for discharging pounds of steam per second at different pressures.

b The maximum amount of steam which the safety valves may be required to discharge when the throttle is suddenly

closed after the fires have been urged to their maximum rate, to be based on the square feet of equated heating surface, so that the relative values for evaporation for various kinds of heating surface, whether of firebox, water tubes for supporting arch brick, long and short boiler tubes between the limits of 10 and 21 ft. in length, and values for different spacing of boiler tubes, will be taken into consideration. Or, what would be simpler, some approximation of average value of heating surface, corrected to account for difference in length and spacing of tubes; the firebox heating surface in this case to be considered as a certain percentage of the whole for all sizes of locomotives.

6 A great diversity of practice exists in the spacing of flues in locomotive boilers. The variation in length ranges in common practice from 10 to 21 ft. These two conditions make the use of heating surface unequated as an absolute guide for the amount of water evaporated, somewhat unreliable. It is evident in two boilers having the same diameter and the same length between flue sheets, that one will contain a much larger amount of heating surface if the flues are spaced $\frac{11}{16}$ in. apart than the other if they were spaced 1 in., and both these figures are within the limits of what is accepted as good practice. Furthermore, heating surface based on flues of the same diameter, and say 11 ft. long, will be much more effective per foot of heating surface than if they were 21 ft. long. Fire box heating surface is, of course, very much more efficient than tube heating surface, and the water-tube heating surface for supporting firebricks is more efficient than the ordinary boiler tubes.

7 Tests show that the evaporation of boilers is somewhat independent of the tube spacing, and probably is more in direct relation to the cubical contents, as it is a matter of common knowledge that the steaming capacity of boilers does not vary in direct proportion to the amount of heating surface if a great variation exists in the spacing of the tubes.

8 The evaporation per square foot of heating surface in locomotives is a variable quantity, ranging from 6 lb. or even less to 15 or 16 lb. per square foot of heating surface per hour. On the authority of Professor Goss, from Purdue University tests, it may be stated that the evaporation in a very general way, and the draft produced by the blower and exhaust in terms of inches of water, will be approximately as follows:

1-in.	draft	will	evaporate	3.0	lb.	per	feet	of heating	surface	per	hour
2-in.	"	"	"	6.0	"	"	"	"	"	"	"
3-in.	"	"	"	8.2	"	"	"	"	"	"	"
4-in.	"	"	"	10.0	"	"	"	"	"	"	"
5-in.	"	"	"	11.4	"	"	"	"	"	"	"
6-in.	"	"	"	12.6	"	"	"	"	"	"	"
7-in.	"	"	"	14.0	"	"	"	"	"	"	"
8-in.	"	"	"	15.0	"	"	"	"	"	"	"

Dr. Chas. E. Lucke. Another element in this safety valve question, of minor importance, perhaps, is the time element. The writer has experimented for many years with rapidly rising and rapidly falling pressures, and believes that increase in pressure in a chamber may go momentarily far beyond what a safety valve is set for. Because this excess is only momentary and measured in fractions of seconds, it should not be considered of no consequence: it is indeed of far more consequence, as a suddenly applied load cannot be resisted by the metal under stress as well as a steady load. If then by any romote series of circumstances the pressure in the boiler suddenly rises, as it may, the time element will enter in, the pressure will go higher than the safety valve is set for, before the valve opens, and will suddenly stress the entire structure. This subject should be studied experimentally, with the others involving the steady rate of steam discharge to discover if it is of any consequence in practical safety valve work. Although the writer has never seen the pressure rise in a steam boiler in this way, he believes it could so rise, producing the effects described.

Jesse M. Smith. Dr. Lucke has touched on a point which needs investigation. Another point along the same line is the danger of having a safety valve too large, particularly if it be of the "pop" kind. If a boiler be stored with water at a temperature corresponding to 150 lb. pressure, and that pressure be suddenly reduced a portion of the water will instantly flash into steam and the boiler may be greatly strained and may explode. There is danger from having a safety valve too large as well as from having it too small. Those who have had to do with the investigation of boiler explosions and particularly those being questioned with regard to them in the courts will realize the necessity for rules based upon scientific investigation and reason, instead of rules having no special reason for their existence except that they have existed for hundreds of years or more. In the courts we are questioned as to rules for safety valves which

have no reason in them, and sometimes we are brought to realize that the safety valves are not in accordance with these rules.

The court may, therefore, hold us liable for damages due to boiler explosions because our safety valves may not be in accordance with some obsolete or erroneous rule adopted by some incompetent state or municipal government.

MR. GARLAND P. ROBINSON. The importance of a careful study of safety valves is, I think, at last fully realized. Although thousands of valves are in daily use, there has been no hitherto satisfactory rule to aid the designer or user in determining the number and size of valves required.

2 The derivation of a formula for the size of valves for stationary and marine boilers would appear to be a comparatively simple problem. Locomotive boilers, however, are operated under entirely different conditions and the determination of the proper size of valves is difficult. The problem in locomotive work appears to lie in the proportion of the maximum evaporative capacity of the boiler to be provided for. Present practice seems to show that it is necessary to provide for about 50 per cent of the maximum evaporation.

3 The commission with which I am connected has collected reliable data on about 7500 locomotive boilers, and during the past week I have calculated the valve capacity of 1000 of these boilers for the purpose of finding the average practice for safety valve equipment. The greatest variations have been noted. For instance, boilers using 180 lb. pressure with valves of $\frac{1}{16}$ in. lift have two 3 in. valves to take care of the evaporation from 1750 to 3350 sq. ft. heating surface, and again we find two $2\frac{1}{2}$ in. valves used to take care of 900 to 1900 sq. ft. heating surface. These cases represent whole classes and not individual boilers. Therefore it would appear that no rule has been followed to determine the size of valve required.

4 The function of a safety valve, as used on a steam boiler, is to discharge steam so rapidly, when the pressure within the boiler reaches a fixed limit, that no important increase of pressure can then occur, however rapidly steam may be made.

5 The heating surface, all things considered, is the best unit of measurement for determining the size of safety valves for locomotive boilers. In my opinion a formula based on the heating surface, and providing for 50 per cent of the maximum evaporation of the boiler, will give satisfactory results for locomotives. A formula for size of safety valves for locomotive boilers can be derived in the manner shown in Mr. Darling's paper on Safety Valve Capacity.

6 For locomotive valves with 45-deg. valve-seats, I would use the formula

$$D = 0.05 \frac{\text{heating surface}}{L \times P}$$

For locomotive values with flat valve-seats, I would use the formula

$$D = 0.033 \frac{\text{heating surface}}{L \times P}$$

7 I have checked 1000 boilers and find the average constant is 0.0441 for present practice. Included in the 1000 boilers, however, are a number which are evidently under-safety-valved, as the constant in their case is only 0.024. Eliminating this class of boilers, the constant for average practice is about 0.05 as given in the formula.

Mr. Wm. H. Boehm. I do not know of any boiler explosion caused by insufficient safety-valve capacity, although I have no doubt that explosions from this cause have occurred. The trouble about a boiler explosion is that after it occurs there is usually not enough of the boiler left to determine the cause of the accident.

2 I think the *time element* has a great deal to do with the safety valve question. If a safety valve is too large it may suddenly relieve so much pressure as to cause a water-hammer sufficient to produce a violent explosion of the boiler. I infer that this is the point the president had in mind when he referred to the danger of using too large a valve.

3 I believe the correct method of determining the size of a safety valve is to take into consideration the actual quantity of steam to be discharged in a given time, rather than to take the heating-surface, grate-surface, etc.

Mr. H. C. McCarty.[1] Safety Valves have passed through various stages of development by the manufacturers, although the subject is not a common one for treatment by the mechanical associations of the country.

2 Reference has been made to the difficulties or objections involved in the use of too large a safety valve. Our experience has been that a valve considered too large is one with excessive lift as compared

[1] Mr. H. C. McCarty, President Coale Muffler and Safety Valve Co., Baltimore, Md.

with the average lift of safety valves made by all reliable manufacturers.

3 Little or no specific reference has been made to the mechanical difficulties arising from valves with unusual discharge capacity produced by increased lift. In actual locomotive service we have found, that increasing the lift beyond that commonly followed as indicated by experience dangerously impairs the life and consequently the reliability of the valve, making it excessively expensive for the railroads to maintain, and unreliable and short-lived as a safety device. The hammer-blows referred to are largely the result of extraordinary lift valves, which not only cause the destruction of the valve in a comparatively short time, but are injurious to the boiler in general.

4 Railroad men in close touch with locomotive maintenance are quite familiar with the expensive renewals of cylinder heads and steam chest covers, and other associated difficulties, many of which are caused largely by water in the cylinders, and any condition contributing to this class of engine failures should be avoided.

5 Our experience has clearly proved that safety valves with *unusual discharge*, resulting from *increased lift* of valve, cause a violent disturbance in the water level, especially on the large modern locomotive boilers, and in proportion to this disturbance is the volume of water passing the throttle valve, and hence to the steam chest and cylinders, increased. Railroads will be relieved of many expensive repairs by reversing these conditions, and thus produce the driest steam possible for the engine. To this end, the throttle valve, as is well known, is located at the highest possible point in the boiler. Further to secure greater locomotive efficiency in this direction, the safety valve should be at as high a point on the boiler as clearance limits permit, and with an independent short connection of ample dimensions to the boiler.

6 Our observations have further been that the location of the valve on a boiler has much to do with the normal crest of the water. Air brake shocks in train and similar effects, in conjunction with high-lift valves, have been a frequent cause of locomotive failures through the combination of undesirable conditions, all of which cause a greater volume of water to pass through the throttle valve and safety valve.

7 In a practical study of the subject, we are confronted with these conditions that should be considered by railroad representatives in any effort to secure the greatest locomotive efficiency, with a minimum of locomotive failures and a consequent low cost of maintenance.

8 In our experience in locomotive service we have never had even a suggestion of the necessity or the advisability of increasing the lift of valve; on the contrary, the reverse condition, from a service standpoint presents itself. The possible limited economy in first cost, of a slightly smaller valve having increased lift, to accomplish increased discharge, compared with the next larger size valve with normal lift, is deceptive, as the short life and expensive maintenance of the high-lift valve make it not only an expensive burden to the railroads, but an unreliable device.

Mr. M. W. Sewall. There seems to be some misunderstanding in regard to the safety valve question, arising from following the old rules of practice without much regard to the requirements. Two items need to be considered; (*a*) how much steam will the boiler make; and (*b*) how much steam will the valve deliver. If these two requirements are adapted to each other no difficulty need arise from the use of a high lift valve. The areas of approach to the valve and discharge from it should then be such as practice has already shown to be essential.

2 The amount of steam that a boiler can make does not depend altogether on the heating surface. The most important consideration is rather the amount of coal which can be burned under the boiler, whether on a small grate or a large one, assuming, of course, that the boilers are fairly efficient.

3 The usual diameters have been mentioned in the discussion as if they could not be changed, and the high and low lifts have been spoken of as related to those diameters. As the high lift valve has a greatly increased discharge capacity, however, it should be reduced in diameter and an entirely new adaptation of diameters to "pounds of steam discharged" should be made. A manufacturer could then adopt any desired combination of diameter and lift and the valves would be rated on the pounds of steam delivered per second.

Mr. George I. Rockwood. I have been interested to hear Dr. Lucke's suggestion that he has made experiments which indicated that steam may rise in the steam boiler suddenly, without opening the safety valve, and to a point which will endanger the safety of the boiler. I think it is now obligatory upon him to state what his experiments are. There are so many boilers under steam.

2 I think we have heard something tonight that is a distinct advance in the general knowledge we have on this subject. Mr. Darling

has given us the results of a series of tests which are of considerable importance, for they show that there are no two makes of valves that have the same rise, and that they differ among themselves up to 300 per cent. Of course, they will differ enormously in their rates of steam discharge, and I think the representatives of the Casualty and Fidelity Insurance Company would be vitally interested in that.

That leads me to say that I do not understand why boiler insurance companies do not conspire together, as the ordinary fire underwriters do, and have a laboratory of their own to find out the conditions which affect the design of safety valves and devices in general that are used about the boiler plant, and then lay the law down to the several manufacturers and refuse to write insurance where those devices are not used. That is the most successful method of securing effective apparatus for fire protection, and I think it is also bound to come in steam boiler protection.

Mr. A. A. Cary. There is a saying that the recognition of a lack of knowledge is equivalent to the possession of it. This discussion has certainly done much to point out how little our stock of knowledge concerning the design and proper operation of safety valves really is. How many engineers are sufficiently posted on this subject to calculate carefully, when drawing boiler specifications, the size and number of safety valves required for the boilers they specify, and then name this carefully obtained information with the requirements of design and operation in their specifications? I am afraid there are very few, notwithstanding the great importance of this apparatus to protect their clients from disaster in their boiler-houses. I fear this information is too often obtained from the catalogues of boiler manufacturers, which may or may not be safe practice.

2 I was particularly interested in Mr. Carhart's discussion relating to safety valve springs, perhaps owing to my many years experience in the manufacture of springs, and I agree with his statements. The spring having the least amount of lift (or compression) is certainly the safest. With a spring having a greater amount of lift, you increase the bending and torsional strains in the wire, thus carrying them nearer, or perhaps beyond, their elastic limit. I have seen many springs of very poor design used in safety valves.

3 At the December 1901 meeting of the Society, I spoke of the proper proportions of springs. The pitch-diameter of a helical spring is measured from the center of the wire on one side of the coil, across its diameter to the center of the wire on the opposite side.

The ratio between this pitch-diameter of the spring and the diameter of the wire composing it should not be less than 5 to 1, but 7 to 1 is a better minimum proportion. A pop-valve spring certainly should not be wound to a smaller proportion than 7 to 1, and with such spring coiled to a smaller ratio I have found a considerable breakage resulting.

4 One matter deserving careful attention in the design of pop-valve springs is the shape of the section of wire used. Somebody's grandfather seems to have started the manufacture of these springs with wire having a square section and no one has had the courage to depart from this undesirable practice. Unquestionably, the best and safest wire for springs is that of round section. The principal stress occurring in the wire of a helical spring is that of torsion, and in a wire of square section we have the greatest fiber stress occurring at the corners of the square, which are the most distantly removed from the center of the section.

5 These corners are really the weakest part of the section owing largely to the heat treatment to which the spring is subjected. In heating the spring, prior to tempering, the corners of the square wire are the first parts of the section to become heated and are the most liable to become over-heated. When the spring is plunged into the cooling fluid, the sharp corners are chilled first, and very suddenly, and are generally found to be harder than the interior of the wire composing the spring. For these reasons, small checks or cracks are liable to be developed in these corners. Any one having had experience in tempering steel dies, rolls, etc., knows the necessity of avoiding sharp corners as starting-places for these incipient cracks which so often result in the destruction of the tempered piece.

6 These statements are not based upon theory alone as I have had ample opportunity to note the *much* larger percentage of breakage in the square than in the round-wire springs. In many cases where customers have complained of trouble from breakage, I have induced them to change from square to round wire and their troubles have ceased.

7 The only advantage gained by the use of square-wire springs is reduction of the space required for a spring having the same resistance to compression, but as the difference in space occupied is not very great, such requirements should not cause the selection of an inferior spring.

8 The most durable of all is the helical spring designed to resist extension, known as an extension spring. When this spring is prop-

erly applied, the load is carried directly along the line of the spring's axis, thus doing away with the "buckling" which so frequently imposes harmful bending strains (in addition to the torsional strain) in the wire composing compression springs. The use of compression springs for pop-valves has become almost universal, but there is no reason why extension springs of good design cannot be used for this purpose. In order to obtain a square bearing at the bottom and top of compression springs, the wire at their ends must be annealed and "hammered down" and then ground flat. After this "squaring-up" process, we frequently find a greater length of wire subjected to torsion on one side of the spring (as the spring is compressed) than on the side diametrically opposite. This frequently causes the harmful buckling and distortion of the spring, and again, the flattened spring ends, by holding the wire rigidly at both ends, cause severe additional stresses in the wire of the spring as explained in my discussion at the 1901 meeting.

9 A certain amount of lag occurs in helical springs just before they begin to compress (or extend), requiring a little greater force to operate than that for which they are adjusted. This fact doubtless contributes somewhat to the sudden "pop" when the safety valve opens; as well as the time-element referred to earlier in the discussion.

10 Every compression spring used in a pop-valve should be so constructed that its coils may be suddenly compressed "coil to coil" many times without showing signs of setting (or shortening, as Mr. Carhart has stated).

11 Great care must be taken in the use of springs applied to valves used on superheaters, to prevent their excessive heating. When the steel is heated to recalescence the spring will collapse and for this reason a pop-valve spring should never be allowed to reach a temperature above 500 deg. fahr., and 450 deg. would be a safer limit.

12 In the Cary process of tempering, invented by my father many years ago, which depended upon heating the spring first to the temperature of recalescence, I found that a straight spring could be bent to almost any odd shape before heating, and after it was raised to this recalescent temperature, it would retain the shape given to it by the former on which it was secured.

Dr. A. D. Risteen.[1] Referring to the suggestion that the different companies interested in boiler insurance—there are probably thirteen or fourteen of them in this country— coöperate to conduct an investigation for their mutual benefit, the idea seems to me a very good one, and this subject of safety valves would be an excellent one to take up. When Mr. Rockwood suggested, however, that we try to lay down the law to the manufacturers and owners of boilers, I think he named a task from which the boldest might shrink.

Mr. F. L. Du Bosque. The discussion has shown us who is responsible for the incorporation in the United States Marine Laws of a formula for deciding the size of safety valves, which has caused marine engineers considerable trouble. No formula on mechanics of any character whatever should be incorporated into a law unless the formula is complete, and without doubt should not be so incorporated when some of its factors are left to the opinion of any one of a great number of persons concerned in its use. The formula mentioned has this very serious defect and therefore should not have been incorporated into the law.

2 The factor W is made up of two quantities, the calorific value of the fuel and the amount burned per square foot of grate surface, and the value of these factors can with reasonable judgment be varied so as to vary the size of the safety valve at least 50 per cent. It is now impossible for a designer to specify the size of a safety valve on a marine boiler without first obtaining from the United States inspector his opinion on the value of these two factors, notwithstanding the fact that the inspector who is compelled to decide this question cannot possibly have as much information to assist him as the designer. On vessels licensed by our Government, that operate on long and continuous routes, the calorific value of fuel, and amount burned per square foot of grate surface, can be determined with a reasonable degree of accuracy, but these vessels are probably not more than 20 per cent of the total number of vessels to which this safety valve formula is to apply. The remaining 80 per cent burn fuel varying at times in calorific value in the ratio of one to two and their fuel consumption per square foot of grate varies in nearly as great a ratio.

3 This new formula, therefore, has not in any way improved

[1] With the Hartford Steam Boiler Inspection and Insurance Co., Hartford, Conn.

the Rules of the Steamboat Inspection Service and, as pointed out above, has added only a complication. As to results produced by it, it is easy to see that by selecting proper proportions for the two factors that make up W, and these factors both may be within reasonable limits, the same result will be obtained as by the old formula. The old formula at least gave the designer a certain basis to work on, and if he was designing his work with the proper regard for safety he had the privilege of deviating from the formula if he felt it did not provide a valve of large enough size.

4 This new formula is also similar for cylindrical and water-tube boilers. Practical operation shows that a safety valve on water-tube boilers should be much smaller than on cylindrical boilers of equal evaporative power. A sudden release of steam pressure in a water-tube boiler with its limited water-line area causes more damage by lifting the water within the boiler than can be caused by a moderate increase in steam pressure.

Mr. L. D. Lovekin. In reply to the remarks by Mr. DuBosque, I was not aware of the trouble I had caused marine engineers, and still further, I see no reason for such trouble. When I design a boiler, I always state on the drawing the rate of evaporation expected and send the plan to the inspectors for approval. I have as yet to have a boiler returned by the inspector for the first time, asking for a change in the sizes of safety valves. I think the inspectors realize that the designer is the one to settle on this.

2 I have had considerable experience with the designing of boilers for different steamships throughout the United States, both for natural and for forced draft, and was therefore aware of the very crude conditions existing, relating to sizes of safety valves for such boilers as are in use by the Steamboat Inspection Service. I discussed the matter fully with a number of engineers and showed them the new formula which I proposed submitting to the Board, and all agreed that my formula was based upon common sense.

3 You all know that any safety valve based on one square inch of safety valve for three square feet of grate area for a Scotch boiler, and one square inch of safety valve for six square feet of grate area for a water-tube boiler, is absurd, and yet this was the formula used by the United States Inspectors for many years.

4 I see no reason for a jump in the size of safety valves from 4.16 to $4\frac{1}{2}$ in. in diameter, as a $4\frac{1}{4}$-in. valve would answer the purpose. When the new rule was adopted, the idea was to limit the size of

safety valves to about $4\frac{1}{2}$-in. in diameter on account of the danger of using large valves, and the understanding was that if a valve came to an odd size for manufacture, we would simply take the next size larger.

5 I firmly believe that the safety valve for any boiler should be based on evaporation only, as otherwise the danger lies in the boiler generating more steam than the safety valve can carry away.

6 Mr. Nelson Foley, of England, who is quite an engineering authority, states that safety valves may be made capable of lifting say $\frac{1}{8}$ of their diameter; that a high lift is useless and may be an evil if anything gives way; that the waste steam pipe, when not under the Board of Trade, may be equal in area to the opening with the lift just mentioned, i.e., the area of the waste steam pipe would be one-half the gross cross-sectional area of the valve.

7 Our United States Navy Steam Engineering Department, with all their experience in connection with boilers, have agreed with several prominent authorities abroad on a lift of $\frac{1}{8}$ the diameter of the valve. It does not follow, however, because a valve has provision for a lift equal to $\frac{1}{8}$ of its diameter, that it ever lifts this amount. It is simply a provision in case the valve is required to be lifted by the safety valve hand-operating gear usually provided on all ships.

8 The area of waste steam pipe on all our recent naval vessels is made $\frac{1}{2}$ the gross cross-sectional area of the valve, which accords with the statements of Mr. Foley.

9 There seems to be a misconception as to what constitutes a high lift. There is probably not a safety valve manufacturer who cares to see a safety valve lift $\frac{1}{4}$ in., no matter how large the valve is, in fact, the writer's rule is based on the proportion of $\frac{1}{32}$ the diameter for the lift, and therefore becomes $\frac{1}{8}$ in. on a 4-in. valve; if worked out below a 4-in. valve, the lift becomes proportionately less, and if used for a $4\frac{1}{2}$-in. valve the lift does not become excessive under the formula.

10 It is a coincidence that while the present rule might give an excessive lift on sizes above $4\frac{1}{2}$-in. diameter, it averages up closely to the sizes recommended by many manufacturers for valves below $4\frac{1}{2}$-in. diameter. The rate of evaporation of 180 lb. in the present rule almost coincides throughout with the Board of Trade formula for safety valves under natural draft.

11 It appears therefore that the Board of Trade thought it wise to keep all boilers under natural draft at the same rate of evaporation, i.e., all boilers worked under natural draft are assumed to be

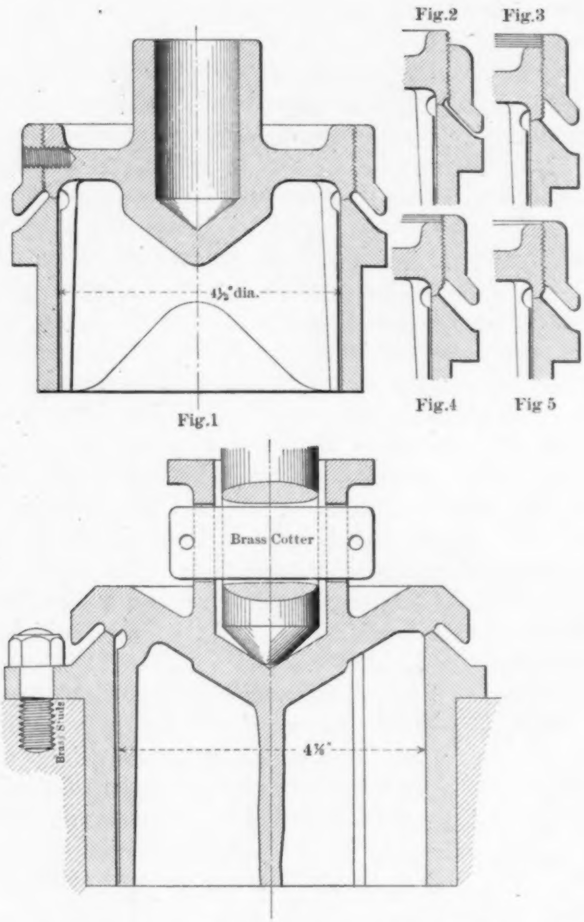

FIG. 1 RESULTS OF EXPERIMENTS ON SAFETY VALVE LIPS

VALVES ROSE AT 81 LB. AND LIFTED ABOUT $\frac{3}{32}$ IN. RATIO OF VALVE AREA TO GRATE AREA $\frac{1}{2}$ SQ. IN. TO 1 SQ. FT. FIG. 1 VALVE CLOSED IN $\frac{1}{2}$ MIN. AT 79 LB., AND VIBRATED CONSIDERABLY. FIG. 2 BLEW STEADILY, WITHOUT CLOSING. FIG. 3 CLOSED IN 1 MIN. AT 80 LB. PRESSURE DROPPED STEADILY. FIG. 4 SAME AS IN FIG. 3. CLOSED IN $\frac{3}{4}$ MIN. AT $79\frac{1}{2}$ LB. LESS VIBRATION THAN IN FIG. 1.

capable of evaporating 180 lb. of water per square foot of grate surface, which seems to be a safe maximum rate for any marine boiler under natural draft.

12 When forced draft is used, under the Board of Trade regulations, the area of the safety valve must not be less than that found by the following formula:

$$A \times \left\{ \begin{array}{l} \text{estimated consumption of coal per} \\ \text{square foot of grate, in pounds per} \\ \text{hour} \div 20 \end{array} \right\} = \begin{array}{c} \text{area of valves} \\ \text{required} \end{array}$$

A equals the nominal area of the valve, based on its diameter, as found from the table of safety valve areas under the Board regulations.

13 The results of experiments on safety valve lips illustrated in the figures may be of interest to the members of the Society. These experiments were made by Mr. Nelson Foley to determine the effect of adjusting the lip on safety valves.

Mr. A. B. Carhart. Certainly there is one way in which safety valves should not be rated, and that is by the area of the disc or of the inlet connection; for in every case the outlet discharge capacity is proportional to the circumference of the valve seat, and the circumference will of course increase in proportion to the diameter, while the inlet and disc areas will increase in proportion to the square of the diameter. If the lift of the disc is the same for all the ordinary sizes of valves, the discharge areas and capacities of the valves are directly proportional to the diameters, and the inlet diameter size becomes a direct measure of the relative size or capacity of the valve. There seems to be no good reason to depart from this method of denoting valve sizes, which has been the uniform custom in the past and will be found more accurate than any other method.

2 The lift may properly be assumed to be uniform in all sizes of valves such as we are considering, for this is the actual performance in practice. Any measurable difference in special cases will generally be found due to the larger valves lifting less vertically than the smaller as is proper in the interests of prompt and quiet action, durability of the valve and safety to the boiler. The smaller valves have less weight of moving parts, less momentum, less load, and springs of more tractable proportions, and may safely lift higher.

3 Neither should valves be rated in discharge area alone. The discharge rating would be different for every pressure and dependent

upon the care in maintaining the uniformity of commercial springs; and in any case would be a theoretical amount arrived at by a formula which might be amended by any designer or salesman to suit the exigencies of contract price or capacity specification. This would introduce a hopeless confusion in odd sizes, besides leaving the engineer at the mercy of the representations or the misrepresentations of selling arguments.

4 The standard sizes, familiar to all engineers, now denote the size of the inlet pipe connection which must be provided in the boiler. For different designs of valves, having different apparent or claimed efficiencies, allowance can be made in the judgment of the engineer. We do not rate iron pipe by discharge capacity or area, but by commercial diameter sizes, and this custom has never been overturned at anyone's suggestion merely because the inside diameter of hydraulic or extra heavy or brass pipe differs from that of ordinary pipe, or because bends and elbows may reduce the flow; engineers exercise their judgment in specification, and this is their proper function and province.

5 The actual lift or discharge areas of valves should be determined and reported upon after impartial tests conducted by competent and disinterested engineers under conditions of scientific accuracy and with due precautions, where each valve is intelligently regulated to work to the limits normally intended; and not from reports of tests conducted by any one manufacturer without the knowledge of other makers whose valves are tested, and where the one measurement noted has been in many cases purposely limited by the manufacturer for special reasons.

6 In rating a valve by its diameter, we use the small diameter of the seat, measuring the area open to the steam pressure when the valve is closed. The seat is made on a comparatively narrow face, preferably not more than $\frac{3}{32}$ in. broad. The line of steam-tight contact is a variable or wavy line, which can be detected by rubbing the beveled face of the valve seat with blue chalk and grinding the disc on it. In no case is it mathematically measured by the inlet diameter; it is always a fractional percentage greater than that, but the difference is ordinarily neglected, and is very small. In very accurate work the mean between the inside and outside diameter would be more nearly correct, but in all commercial ratings, the nominal diameter of the throat is taken.

7 The actual area is determined by experiment and by allowances arrived at by experience. That is, a new valve might have the seat

near the inside and an old one at the outside edge of the seat face, and when it is properly drawn it will be seated either one way or the other. But this would not affect the calculation of the amount of steam that would flow through the valve opening, for that is determined by the circumference of the inner edge of the seat face, where the flow of steam comes up through the throat of the valve and turns outward over the edge of the seat. This sinuous line of contact on the seat face when the valve is closed would affect only the total spring load required or the pressure upon the disc by the confined steam when the valve is closed, because of the slightly varying area open to the steam; but I believe this does not affect the rate of flow or the amount of the opening, as soon as the valve lifts.

Mr. Nathan B. Payne. The most important point brought out by this discussion is that there is no proper standard of measurement for the safety valve's capacity. Whether we take a high-lift or a low-lift valve, what we must have is some way of measuring what relieving capacity we are getting. When we buy a 4-in. valve, for instance, we want to know whether that valve has relieving capacity for a 100-h.p. or a 200-h.p. boiler, or what size it is suited for. Here is a magnificent opportunity for this Research Committee to take up this subject and adopt some standard method of determining the capacity of safety valves.

2 We have been going ahead thinking we were right on the relieving capacity of safety valves when considering only one dimension, but it is absolutely impossible to determine the amount of relieving capacity in a given time without knowing the lift of the valve off the seat, so as to get the effective area of opening. The question for the user to decide is how much relief he can get from a given make and size of valve. If one maker offers a safety valve having $\frac{1}{32}$ in. lift, and another offers a $\frac{1}{8}$-in. lift, each should state how many pounds of steam per hour his valve will relieve.

Mr. Frank Creelman. I wish to speak regarding Mr. Rockwood's question, as to the insurance companies taking up the matter of the safety valve. The earliest experiments on the safety of parts of the boiler were made for the Manchester Union Steam Users' Association by Sir William Fairbairn, on the strength of boiler flues to resist collapse. This Association's engineers up to this day have continued to carry on original experimental work relating to the safety of boilers.

Mr. Boehm. In speaking earlier of the danger of having too large a valve, and the water-hammer it might cause, the writer had in mind, not too large a diameter but a valve of too large capacity. A small valve with a high lift, if the full effect of that lift suddenly be obtained, might discharge as much steam in the same length of time as a large valve with a low lift. The *time element* was the thing I particularly wanted to mention.

2 Mr. Rockwood had something to say regarding the writer's statement that he had known of no boiler explosion caused by too small a safety valve. The writer did not mean to say that no boiler had exploded for that reason, but that he did not *know* of one. He does know of cases where slice bars and other things were hung on the safety valve lever, of cases where stop valves had been placed between the safety valve and the boiler, and of one case in particular where an engineer (?) in New York had wedged a piece of scantling between the joist and the top of the safety valve lever.

Mr. Pond. The paper read by Mr. Darling is particularly interesting as indicating what has already been done in the direction of obtaining reliable data relative to the operation of safety valves, from test.

2 I agree with Mr. Ashton that the lift of the valve is not the essential thing. The thing to be determined is how much steam a given valve will discharge under any given set of conditions. That particular piece of information is the one that none of the valve manufacturers have been able or willing to give us, as none of them have previously made the tests necessary to determine these points. This information relative to valve capacity is what I have been trying to get and I have every reason to believe that very shortly we will have presented results of a number of tests bearing on this subject. Tests are being conducted and being prepared for at the present time, from the data of which we may hope to determine more accurately the correct proportions of safety valves for a given service.

3 This question of high-lift and low-lift valves seems to me to be one simply of capacity. If the low-lift valve will deliver a certain number of pounds of steam at a given pressure and temperature, and we know what its capacity is under these conditions, this is the principal thing required. The same test applies in the case of the high-lift valve, the essential point, however, being to know how much steam the valve will discharge. Undoubtedly a high-lift valve will give more capacity than a low-lift valve having the same diameter of open-

ing. This being so, we could use a smaller valve of the high-lift type on a boiler, than of the low-lift type, which would be an advantage in many ways.

4 I trust that before this subject is left steps may be taken by the Society to take definite action on the formulating, or rather revising, of the safety valve rules for stationary, locomotive, and marine practice, standardizing them on the basis of the capacities of the valves.

5 The first step is to get the manufacturers or some disinterested parties to make quantitative tests which will give us an accurate measure of how much steam can be discharged through a safety valve.

MISCELLANEOUS DISCUSSION

LIQUID TACHOMETERS

By Amasa Trowbridge, Published in The Journal for Mid-October

The Author. Mr. Moss is correct in his statement regarding the accuracy of a liquid tachometer. The general impression seems to be that the instrument wears in such a way as to effect its readings. Such is not the case. Any leak caused by wear would render the instrument inaccurate until the leak was stopped, but as soon as this trouble was remedied the instrument would again be accurate without being recalibrated.

2 In regard to building an instrument with a vertical driving-shaft, this is a perfectly feasible proposition. The opening for the shaft would have to be kept above the zero level of the instrument to avoid the necessity of a stuffing-box. If this were done, wear would not make the instrument leak. It is probable that an instrument of this form will be put on the market as soon as there is sufficient demand for it.

3 In most cases the instruments are not used for continuous running, but are applied to continuous running machines in such a way that they can be thrown into or out of engagement at will, and the speeds are usually indicated only at stated intervals. In this way the wear on the instrument is slight and trouble is not experienced from leaking through the stuffing-box.

THE TOTAL HEAT OF SATURATED STEAM

By Dr. Harvey N. Davis, Published in The Journal for November

The Author. The first seven paragraphs of Professor Thomas' discussion are a valuable contribution to the outstanding C_p controversy, especially his statement that according to his new experiments, "there is no question that a comparatively very large amount of heat is required to cause a very small rise of temperature of dry saturated steam." The publication of these new results will be eagerly awaited by all interested in the subject. In the meantime, it should be remembered that this C_p controversy has much less to do with the

validity of the results in this paper than might at first be supposed. As has already been pointed out in Par. 3 of my closure, the use of Professor Thomas' values of C_p instead of Professor Knoblauch's would make only a small difference in H. It is interesting to notice that the changed values of H would be even farther from Regnault's than are those proposed in this paper.

2 The last three paragraphs of the discussion raise a much more vital question as to the validity of the throttling experiments on which this paper is based. This criticism has also been made by Professor Heck. It is true that throttling experiments have fallen into disrepute "in view of the well-known troubles that have been experienced" in "two lines of experimental work," namely in determining the quality of wet steam and in computing C_p from H. Of these, the latter is a use for which such experiments are particularly ill-adapted, and it is this very fact which makes the reversal of the process—the computation of H from C_p—so insensitive to errors in C_p. As to the former, one should remember that the ordinary throttling calorimeter, even when "great care is employed as to lagging, position, etc.," is the crudest sort of an instrument of precision, as far as heat insulation and the measurement of the low-side temperature are concerned, so that it is not remarkable that great accuracy is not attained.

3 The experimenters whose results are the basis of this paper used apparatus of a very different sort. Their three different systems of heat-insulation and of thermometry, although by no means perfect, were much better than those of which the average engineer would be reminded by Professor Thomas' allusions to throttling calorimetry. If the precautions of any one of them had not been effective, no such agreement of results based on their work could possibly have been expected as is actually found. The value of their mutual corroboration, in every respect that concerns this paper, is increased by the fact that both Griessmann and Peake were primarily interested in disproving a certain conclusion of Grindley's, so that their critical attitude might have been counted on to ensure substantial improvements in Grindley's results, if that had been possible. If this paper had been founded on Grindley's work alone, the doubts of Professors Thomas and Heck would have had great weight; but the fact that all three pieces of work were used, *and the results agreed,* is good evidence that these throttling experiments are beyond the uncertain and inconsistent stage which Professor Thomas describes.

Prof. Carl C. Thomas. The discussion during the past few months regarding the properties of superheated steam, and also that which now bids fair to throw additional light upon the total heat of saturated steam, has been exceedingly valuable. The discussion has involved the work of a number of experimenters, and a few points concerning the results of some of the experiments ought to be emphasized lest they be lost sight of or misunderstood. Also, the writer wishes to make a suggestion regarding the proposed revision of the steam tables upon the basis of throttling calorimeter experiments.

2 Knoblauch's experiments begin at about 30 to 50 deg. fahr. superheat for each of the four pressures used; the writer's begin at 18 deg. fahr. superheat for all the pressures used Knoblauch's upper limit of temperature varies from about 325 to 400 deg. fahr. superheat, while the writer's experiments stop, in all cases, at 270 deg. fahr. superheat. In that part of the temperature range which is common to both sets of experiments, namely, between 30 or 50 deg. fahr. and 270 deg. fahr. superheat, the results in the two cases are almost identical. The greatest variation appears at about 28 lb. absolute pressure and 54 deg. fahr. superheat where Knoblauch obtains $C_p = 0.478$ and the writer obtains 0.498. The writer's experiments from 18 to 50 deg. fahr. superheat show higher values of C_p than are shown by the extrapolated curves of Knoblauch.

3 Knoblauch worked at absolute pressures of about 28 lb. to 114 lb. while the writer worked at absolute pressures from 7 lb. to 500 lb. Knoblauch obtained his saturation curve by extrapolation from experimental determinations at about 30 or 50 deg. fahr. superheat, while the writer obtained his saturation curve by extrapolation from exper ments made at 18 deg. fahr. superheat. The two curves show widely different values for C_p at saturation, the writer's being much higher than Knoblauch's.

4 In view of the very close agreement in values of C_p from 50 to 270 deg. fahr. superheat, obtained in these two sets of experiments made by entirely different methods, it seems safe to accept either set of these values as substantially correct for this temperature range.

5 Correspondence which has passed between Professors Schröter and Knoblauch and the writer brings out the fact that the former are pushing their experiments into the higher temperature ranges, desiring to corroborate and extend their already published results, which show a minimum value of C_p somewhere in the neighborhood of 230 deg. fahr. superheat for each pressure, followed by an increase of

C_p after the region of minimum value has been passed. Their method and apparatus are especially well adapted to experimenting with highly superheated steam. On the other hand, the writer is working back towards the saturation condition, with an apparatus which has been planned with the special end in view of definitely locating the saturation curve. Appreciation of the real difficulties in obtaining exact knowledge as to moisture conditions existing in steam can be attained only by those who have made actual and extensive experiments.

6 The writer's experiments have led him to expect to find a saturation line, and not a "saturated region." Upon this assumption, such curves as are shown in the writer's Fig. 13 should, as shown, pass through the intersection of the coördinates, namely, heat introduced per unit weight of steam, and temperature to which the steam is being superheated. It is with the determination of the form of these curves, near the intersection of the coördinates, that the writer is now engaged. There is no question that a comparatively very large amount of heat is required to cause a very small rise of temperature of dry saturated steam, which means that C_p is comparatively large in value right at saturation. The writer is attempting to ascertain, by the use of very sensitive resistance thermometers instead of the thermo-couples formerly used, whether it is a case of "jogging the steam out of the saturation region," as some writers have suggested, by the introduction of a considerable amount of heat—or whether any small increment of heat will cause a correspondingly very small rise of temperature of steam which has just reached the condition of complete dryness.

7 In the absence of further data at present which would tend to establish the relative correctness of Knoblauch's and the writer's saturation curves, it is interesting to notice that curves can be drawn through Knoblauch's points as given in his Fig. 12 which will produce the writer's guess at the saturation curve, quite as readily as the one Dr. Knoblauch has made. And, on the other hand, it would be quite possible to use the writer's data as an argument in favor of the accuracy of Knoblauch's curve. However, the writer has some confidence in the general reliability of his curve, because it is based upon experiments extending down to within 18 deg. fahr. of the saturation temperature, while Knoblauch stopped his experiments at a considerably higher temperature. The fact that the two sets of results agree so well, within the temperature-range actually covered in common by the two sets of experiments, affords evidence of the reliability of the values

given for C_p; and the further consideration that each of the writer's experiments started with a determination of the saturation point as its basis, and from that went up and fairly met Dr. Knoblauch's results, which were obtained without reference to the saturation point, gives reason for accepting with some confidence the saturation curve marked out by the writer's experiments. However, it will be apparent from the above remarks that the writer is going over this whole question of the saturation line again in the experiments now in progress.

8 The reason why it is worth while to go carefully into this matter is apparent, and is especially cogent at the present time from the standpoint of the engineer as well as the physicist, because of the interesting suggestion of Dr. Davis that the steam tables based on Regnault's classic experiments be revised on the basis of experiments which have been made on the expansion of steam in throttling calorimeters, or through porous plugs, for the purpose of determining the specific heat of superheated steam.

9 It is not the writer's intention or desire to detract from the interest attaching to Dr. Davis' proposed use of throttling experiments for indicating the extent to which Regnault's values of total heat of saturated steam should be changed. The indications obtained from the throttling experiments will undoubtedly be suggestive and possibly conclusive, but they should not be accepted, without experimental corroboration, as a basis for a revision of the steam tables. During the past eighteen years, to the writer's knowledge, experiments have been made with the object of obtaining values of C_p by throttling steam. The results ha e never been consistent among themselves, nor have they given values for C_p comparable with those which have been made by the direct means of electrical heating of the steam. For a time some of the throttling experiments were accepted by engineers, and it was then thought that the specific heat of superheated steam was very much higher than it has since been shown to be, the errors in the throttling method all tending to increase the apparent value of C_p.

10 Useful as the throttling calorimeter is, it is not necessary to remind engineers of the very great difficulty of obtaining reliable or at all consistent results as to quality of steam, by its use, no matter how great care is employed as to lagging, position, etc. It may be a question of uniformity of steam conditions at inlet to the instrument, or in the instrument itself at the point where the temperature is taken. The writer is aware that the method by which Dr. Davis

proposes to use the results of calorimeter experiments does not involve precisely the same considerations as those involved in determining the quality of steam, or in fact of determining the specific heat of steam by the throttling method. But in view of the well-known troubles that have been experienced in these two lines of experimental work, it seems decidedly inadvisable to base such an important matter as a revision of the steam tables upon experiments in throttling steam, without thorough corroboration by direct measurement.

11 The direct measurement of the total heat of dry steam will be possible of attainment with great accuracy as soon as the saturation line above referred to, and the heat necessary to raise the temperature of unit weight of steam some distance above saturation, have been definitely and finally determined. Means of applying heat electrically, in absolutely measurable quantity sufficient not only to evaporate water into dry steam but to superheat the steam by some known amount, afford means of direct measurement of the total heat of dry steam, because the heat applied above that necessary to obtain the saturation condition will be known, as soon as the saturation curve has been determined, and this superheat can be subtracted, leaving as a remainder the total heat of dry saturated steam.

12 The writer would not wish it understood that he urges delay in revising the steam tables until his own experiments shall have been completed. That is not at all the point. By the time the saturation curve is definitely located, and possibly before, the writer will be glad to turn the work over to anyone else who may be in position to go on with it; or the desired results may be obtained by some other experimenter entirely independently and before the writer's experiments can be brought to a satisfactory conclusion; but since methods of direct electrical measurement are now available and admirably adapted to this purpose, such means should certainly be used as a check upon the accuracy of the methods proposed by Dr. Davis, before engineers are asked to accept a new set of steam tables. Otherwise we shall have just such a condition of uncertainty regarding the accuracy of our steam tables as has existed for many years with regard to the specific heat of superheated steam.

ACCESSIONS TO THE LIBRARY

BOOKS

Annual Report of the Boston Transit Commission. 14th. *Boston, E. W. Doyle, 1909.* Gift.

Annual Report of the Light-House Board to the Secretary of Commerce and Labor, June 30, 1908. *Washington, Government, 1908.* Gift.

Annual Report of the New York State Museum. 61st. Vol. 1 and 3. *Albany, University of the State of New York, 1908.* Gift.

Bulletin of Syracuse University. Series 8, no. 4. *Syracuse, 1909.* Gift.

Cassier's Magazine. Vol. 1, 1891–1892. *New York, 1892.* Gift of H. H. Suplee.

Cuyahoga Valley Viaduct of the "Nickel Plate" Railroad. Read at the Meeting of the Cleveland Engineering Society, October 13, 1908. By G. H. Tinker. *Cleveland, 1908.* Gift of Cleveland Engineering Society.

Development of the Electric Piano Player. By J. F. Kelly. Reprinted from Journal of the Franklin Institute, *Jan., 1909.* Gift of author.

Engineering Magazine. Vol. 1, no. 5 and 6; vol. 2, no. 1 and 2. *Aug.–Nov., 1891.* Gift of H. H. Suplee.

Flight Velocity. By A. Samuelson. *London, Spon, 1906.* Purchase. $0.75.

Flying Machines: Past, Present and Future. By A. W. Marshall and Henry Greenly. *London, P. Marshall & Co.* Purchase. $0.40.

Gas Engineer's Laboratory Handbook. 2d edition. By John Hornby. *New York, Spon & Chamberlain, 1902.* Purchase. $2.

Hobart College Bulletin. President's Report, 1908–1909. *Geneva, N. Y., College, 1909.* Gift.

Mercurial Air Pump, The. By Prof. Silvanus P. Thompson. (Reprint with additions from the Journal of the Society of Arts.) *London, E. & F. N. Spon, 1888.* Purchase. $0.60.

Modern Foundry Practice. 2d edition. By John Sharp *New York, Spon & Chamberlain, 1905.* Purchase. $8.

Portland Cement: its Manufacture, Testing and Use. 2d edition. By D. B. Butler. *New York, Spon & Chamberlain, 1905.* Purchase. $5.

PORTRAIT OF FREDERIC A. C. PERRINE, with a few words regarding his life and work. *1908.* Gift of F. V. T. Lee.

PROCEEDINGS OF THE 18TH ANNUAL CONVENTION OF THE AMERICAN RAILWAY AND BUILDING ASSOCIATION, Washington, Oct. 20–22, 1908. *Concord, N. H., Rumford Printing Co., 1908.* Gift.

PROCEEDINGS OF THE INCORPORATED INSTITUTION OF AUTOMOBILE ENGINEERS FOR 1906–1907. *London, 1907.* Exchange.

REPORT ON THE KANOWNA MINES. By A. Montgomery. *Perth, 1908.* Gift of Western Australia Department of Mines.

REPORT ON THE NORTHAMPTON MINERAL FIELD. By A. Montgomery. *Perth, 1908.* Gift of Western Australia Department of Mines.

RESISTANCE OF AIR AND THE QUESTION OF FLYING. By A. Samuelson. *London, Spon, 1905.* Purchase. $0.75.

SÄCHSISCHER DAMPSKESSE REVISIONS VEREIN, CHEMNITZ. Ingenieur-Bericht, 1908. *Chemnitz, W. Adam, 1908.* Gift.

STEAMSHIP COEFFICIENTS, SPEEDS AND POWERS. By C. F. A. Fyfe. *New York, Spon & Chamberlain, 1907.* Purchase. $4.

DIE TURBINE. Zeitschrift für Moderne Schnellbetrieb für Damp-Gas Wind and Wasserturbinen. Organ der Turbinentechnischen Gesellschaft. Semi-weekly. Vol. 1–4. *Berlin, M. Krayn, 1904–1907.* Gift of Prof. Edward C. Schmidt.

WASHING AND COKING TESTS OF COAL AT THE FUEL-TESTING PLANT. Denver, Colo., July 1, 1907–June 30, 1908. By A. W. Belden, C. R. Delamater, and J. W. Groves. *Washington, Government, 1909.* (U. S. Geol. Survey. Bulletin no. 368.) Gift.

CATALOGUES

AMERICAN IRON AND STEEL MANUFACTURING COMPANY, *Lebanon, Pa.* Boiler Rivets.

Catalogue of Manufactures. *1908.* Refined bar iron, bolts, nuts, rivets, spikes, and forgings for railroad telegraph and telephone work.

CRANE COMPANY, *Chicago, Ill., 1908.* Effect of Superheated Steam on Valves and Fittings.

ECK DYNAMO AND MOTOR COMPANY, *Belleville, N. J.* Sectional Catalogue and Data Book. (Bulletin no. 39.)

GREENE, TWEED AND COMPANY, *109 Duane St., New York, 1908.* Belt studs, hooks, tools, clamps, etc.

NATIONAL ELECTRIC LAMP ASSOCIATION. Tungsten Multiple Lamps. (Bulletin no. 6D.)

Niles-Bement-Pond Company, *Cincinnati, O.* Reprints regarding cutter grinders.

Schutte and Koerting Company, *Philadelphia, and 50 Church St., New York.* Apparatus for: Feeding boilers; Lifting liquids; Heating liquids; Moving air and gases; Atomizing liquids; Valves; Condensers; Hydraulic machinery; Sulphur furnace plants for sulphuric acid manufacturing. Subject index; code words of sizes in inches.

Steam Power Central Station, A Modern. Designed and Installed by Charles C. Moore & Co., *San Francisco, Cal.*

H. N. Strait Manufacturing Company, The. *Kansas City, Kan.* Scales, cooperage machinery, friction clutches, transmission and conveying machinery, etc.

Hubert H. Ward and Associates, *Cleveland, O.* The Seaton Spring Wheel.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

055. Wanted: an assistant superintendent in factory manufacturing a line requiring a large production of small duplicate parts. A man with experience in clock, lock or typewriter factory, with technical training preferred. Applicant must have held executive positions. Location Pennsylvania.

056. A man with shop experience, capable of selling high-grade machine tools wanted, in the middle West. Technical graduate preferred.

057. Designing draftsman and detailers wanted by steel company. Location Pennsylvania.

MEN AVAILABLE

231 Mechanical and Structural Engineer, with business training, going to Europe this summer, would like to act as representative for responsible concern expecting to introduce their goods abroad.

232 Member, 20 years experience with large companies, in design, manufacture and operation of steam, electric and gas machinery, accustomed to handling men and conducting correspondence, desires to get into communication with concern with view to engagement as Assistant Chief Engineer or Chief Draftsman. Best of reference.

233 Junior, technical graduate, going abroad again this June, would like to make the acquaintance of parties having European connections with view to temporary engagement.

234 Member, Stevens graduate, desires to make a change. Competent as Manager, Superintendent or Chief Engineer. Fifteen years experience in machine tools, hydraulic machinery, coal and gas fired furnaces, conveying machinery and plant equipment.

235 Mechanical Engineer, 35 years of age, with practical shop and technical experience in design and construction of special and automatic machinery, manufacturers' tools, for speedy and economical production of interchangeable parts;

thoroughly familiar with best up-to-date shop practice and management, costs and efficiency, good organizer and systematizer with excellent executive ability, and can produce results, desires to locate permanently with concern in vicinity of New York.

236 Superintendent desires position, preferably in manufacturing plant with machine shop and foundry producing medium weight machinery.

237 Member with long experience in design of material-handling machinery, including elevators, conveyors, coal tipples and screens, crushing and concentrating plants, also steel and timber mill buildings, desires to become connected with some responsible concern where his special training will be of value.

238 Junior Member, B.S. and M.E., experienced in marine engineering as draftsman and inspecting draftsman, and in the installation of machinery in battleships and armored cruisers; was evening instructor in steam engineering five years, including laboratory work; at present in charge of a marine engine drawing room; desires position with consulting engineer engaged in power station work or with engineering building company.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ALEXANDER, Chas. A. (1899; 1905), Engr. and Contr., Builders' Exchange, and 3 Cornell St., Rochester, N. Y.

AUE, Jos. E. (1899), Pacific Gas and Elec. Co., Bayshore Dist., San Francisco, Cal.

BARNES, Charles B. (1905: 1908), Mech. Engr., Holabird & Riche, Architects, 1618 Monadnock Bldg., and *for mail,* 325 E. 53d St., Chicago, Ill.

BAUSH, George Henry (1905), M. E. and Genl. Sales Mgr., Fay Mch. Tool Co., and *for mail,* 2101 Spring Garden St., Philadelphia, Pa.

BERNHARD, Richard (Junior, 1901), Ch. Engr., Mining, Crushing and Cement Mchy. Dept., Power and Mining Mchy. Co., Cudahy, and *for mail,* 632 Wentworth Ave., Milwaukee, Wis.

BURTON, J. Harry (Junior, 1906), 1001 W. 71st St., Chicago, Ill.

COES, Harold V. O. (Junior, 1907), Liquid Carbonic Co., Michigan Ave., Chicago, Ill.

COFFIN, Frank M. (1907), Supt. of Constr. and Repairs, The Maintenance Co., 54–56 Franklin St., and *for mail,* 272 Manhattan Ave., New York, N. Y.

COFFIN, Wm. Carey (1894), Structural Engr., Jones & Laughlin Steel Co., and *for mail,* 5930 Howe St., Pittsburg, Pa.

COLE, Cyrus L. (Junior, 1908), Allis-Chalmers Co., and *for mail,* 2010 Kenmore Ave., Chicago, Ill.

CRANSTON, Raymond E. (Junior, 1907), Asst. Engr., 815 Banigan Bldg., Providence, and 31 Lawn Ave., Pawtucket, R. I.

FERRIER, Walter (1908), Carnegie Steel Co., Schoen Steel Wheel Plant, McKees Rock, Pa.

FILLINGHAM, Myles Percy (1908), Cons. and Contr. Engr., Hudson Terminal Bldg., 50 Church St., New York, N. Y.

FREDERICK, Floyd Willis (1907), Mech. Engr., Natl. Board of Fire Underwriters, and *for mail,* 235 Valley Ave., Easton, Pa.

FUCHS, Hugo (Associate, 1907), Cons. Engr., V Lipot Blvd., 27, Budapest Hungary.

GILMAN, Francis L. (1908), Shop Supt., Western Elec. Co., 463 West St., New York, N. Y., and *for mail,* 49 Christopher St., Montclair, N. J.

GWILLIAM, Geo. T. (1891), C. E., Resident Mgr., The Hess-Bright Mfg. Co., 1974 Broadway, New York, N. Y., and The Union League, Philadelphia, Pa.

HARDING, Adalbert (Junior, 1898), The Wickes Boiler Co., Room 1411, 90 West St., and 69 W. 49th St., New York, N. Y.

HART, Rogers Bonnell (Associate, 1907), Ch. Draftsman, Pittsburg Gage and Supply Co., 30th and Liberty Sts., and *for mail,* 705 College Ave., Pittsburg, Pa.

HENES, Louis G. (Junior, 1903), Ry., Industrial and Contractors' Equipment, Room 731, Monadnock Bldg., San Francisco, and Key Route Hotel, Oakland, Cal.

HERR, Herbert T. (1907), Genl. Mgr., Westinghouse Mch. Co., East Pittsburg, Pa.

HILL, Reuben (1908), Factory Mgr., Bristol Engrg. Corp., Bristol, Conn.

HOLLOWAY, Thurman Welford (Junior, 1906), Asst. Prin., School of Mech. Engrg., Internatl. Correspondence Schools, and 1642 Madison Ave., Scranton, Pa.

KENT, William (1880), Manager, 1885–1888; V. P. 1888–1890; Cons. Engr., Sandusky, O.

LANE, Henry Marquette (1900), Editor, Castings and Wood Craft, Caxton Bldg., and *for mail,* 15 The Melrose, 1924 Prospect Ave., Cleveland, O.

LAWRENCE, Howard F. (Junior, 1908), Am. Soc. Mech. Engrs., 29 West 39th St., New York, N. Y.

LOGAN, Wm. J. (1880), Life Member; Logan Iron Wks., Commercial and Clay Sts., Brooklyn, N. Y.

LORD, John E. (1904), Asst. Mgr., Sight Feed Oil Pump Co., and *for mail,* 902 Richards St., Milwaukee, Wis.

MASURY, Alfred Fellows (Junior, 1904), Mech. Engr., Hewitt Motor-Truck Co., and *for mail,* 44 W. 25th St. New York, N. Y.

MICHEL, Arthur Eugene (1906; Associate, 1908), Adv. Engr., 1572 Hudson Terminal Bldg., New York, N. Y.

MILLS, Edmund (1903), 319 Arlington Ave., Jersey City, N. J.

MOLÉ, Harvey E. (1901), Lenz & Molé, 71 Broadway, New York, N. Y.

MORA, Rafael de la (Junior, 1897), Mech. and Hyd. Engr., Hidalgo 654, Guadalajara, Mex.

MURRAY, George R. (1903), Pres., The Maxwell Rolf Stone Co., 586 The Arcade, Cleveland, O., and *for mail,* 129 Pearson Drive, Asheville, N. C.

NEELY, Frank H. (Junior, 1908), Industrial Engr., 70 Madison Ave., Atlanta, Ga.

NORBOM, John O. (1900), Alta Vista Apts., Berkeley, Cal.

PAINE, Henry E. (Junior, 1906), 505 Homer Ave., Palo Alto, Cal.

PARSONS, William N. (1901), 216 Falconer St., North Tonawanda, N. Y.

PRINCE, Walter F. (1905), Supt. Fdy. Dept., Henry R. Worthington, Harrison, and 1365 North Ave., Elizabeth, N. J.

RATHBUN, Edward (1908), 2d V. P., Rathbun-Jones Engrg. Co., Spencer St., Toledo, O.

REDWOOD, Iltyd I. (1890; 1903), Genl. Wks. Mgr., Borax Consolidated (Ltd.), 16 Eastcheap, London, E. C., and *for mail,* 21 Erith Road, Belvedere, Kent, England.

RUST, William F. (1904), Youngstown Sheet and Tube Co., Youngstown, O.

SALMON, Frederick W. (1900–1904), Civil and Mech. Engr., Murray Iron Wks., and *for mail,* 815 N. 6th St., Burlington, Ia.

SHELDON, Samuel B. (Junior, 1900), Genl. Supt., Saucon Dept., Bethlehem Steel Co., S. Bethlehem, Pa.

SHIRLEY, Robert (1906), Mech. Engr., The Pratt & Cady Co., and *for mail,* 26 Lenox St., Hartford, Conn.

SINCLAIR, Angus (1883), Editor and Publisher, 114 Liberty St., New York, N. Y., and *for mail,* 400 Clinton Ave., Newark, N. J.

SMITH, S. H. (Associate, 1907), Supt., North Melbourne Elec. Tramways and Lighting Co., Ltd., Mt. Alexander Road, Ascot Vale, and Clydehall, Harding & East Sts., Melbourne, Australia.

SMITH, William E. (Junior, 1908), Mech. Engrg. Dept., D. L. & W. R. R., and *for mail*, 518 Olive St., Scranton, Pa.

SPURLING, O. C. (1907), Plant Engr., Western Elec. Co., Hawthorne, Ill.

STRAW, Charles A. (1896; 1904), Sales Mgr., Lehigh Coal and Navigation Co., and *for mail*, Lansford, Pa.

STUCKI, Arnold (1907), Engr., Farmers Bank Bldg., and *for mail*, 105 Falk Ave., N. S., Pittsburg, Pa.

SZE, S. C. Thomas (Junior 1905), Imperial Railways of North China, Tongshau, China.

TALCOTT, Robert Barnard (1907), Asst. Ch. Mech. and Elec. Engr., Office of Supervising Arch., Treasury Dept., and *for mail*, Florence Court, Washington, D. C.

THULLEN, L. H. (1905), Mech. and Elec. Engr., 540 W. 143d St., New York, N. Y.

VAN DEINSE, A. F. (Junior, 1905), El Tiro Copper Co., El Tiro, Pima Co., Ariz.

WEST, Arthur (1902), Mgr. Power Dept., Bethlehem Steel Co., S. Bethlehem, and 114 S. High St., Bethlehem, Pa.

WHEELER, Wm. Trimble (1905), Genl. Mgr., Trinity Engrg. Co., 90 West St., and 340 W. 21st St., New York, N. Y.

WHITE, John Culberton (Junior, 1906), Cons. Steam Engr., 745 E. Johnson St., Madison, Wis.

WHITTED, Thomas B. (1900; Associate, 1903), Pres., Thomas B. Whitted & Co., Contr. Engrs., Piedmont Bldg., and 317 W. 5th St., Charlotte, N. C.

WINTER, Oscar (1906), 1360 W. 112th St., Cleveland, O.

NEW MEMBERS

BRONSON, Edward L. (Associate, 1908), M. M., The Shoe Hardware Co., and *for mail*, Montgomery St., Waterbury, Conn.

DUNLAP, Thaddeus C. (1908), V. P. and Genl. Mgr., Columbus Pneumatic Tool Co., Columbus, O.

FARRELL, Harry C. (1908), Mech. Engr., United Shoe Mchy. Co., Beverly, and *for mail*, 47 Grant Road, Swampscott, Mass.

GORE, Warren W. (1908), V. P., Gas Power Mfg. Co., Seattle, and *for mail*, 1610 Main St., Olympia, Washington.

HARVEY, Minor (1908), Engr., Morse, Williams & Co., and *for mail*, 4217 Haverford Ave., Philadelphia, Pa.

HOLMES, Joseph Austin (1908), Expert in Charge Technologic Branch, U. S. Geological Survey, Washington, D. C.

HOWARTH, Harry A. S. (1908), P. O. Box 174, S. Bethlehem, Pa.

KEMBLE, Parker H. (1908), Bristol, Conn.

MONKS, Wm. Douglas (Junior, 1908), 353 S. 3d Ave., Mt. Vernon, N. Y.

SHODRON, John Geo. (Junior, 1908), 220 Burrell St., Milwaukee, Wis.

STAUBE, Edwin G. (Junior, 1908), Pres. and Managing Dir., E. G. Staube Mfg. Co., 526 S. 5th St., Minneapolis, Minn.

PROMOTIONS

WOODWELL, Julian E. (1900, 1908), Cons. Engr., Terminal Bldg., Park Ave. and 41st St., New York, N. Y.

DEATHS

REYNOLDS, Edwin.

BOYER, Francis H.

GAS POWER SECTION

CHANGES OF ADDRESS

AUE, Jos. E. (1908), Pacific Gas and Electric Co., Bayshore Dist., San Francisco, Cal.

COES, Harold V. O. (1908), Liquid Carbonic Co., Michigan Ave., Chicago, Ill.

THULLEN, L. H. (1908), Mech. and Elec. Engr., 540 W. 143d St., New York, N. Y.

NEW MEMBERS

AMSLER, W. O. (1909), Engr., 627 Wabash Bldg., and 5510 Margaretta St., Pittsburg, Pa.

BIRD, John D. (1909), Asst. Genl. Mgr., H. R. Worthington, Harrison, and *for mail*, 100 S. Arlington Ave., East Orange, N. J.

BLAKE, Frederick W. (Affiliate, 1909), Genl. Mgr., United Rys. of Yucatan, and *for mail*, P. O. Box 289, Merida, Yucatan, Mex.

CAHILL, Chas. Adams (Affiliate, 1909), 316 Public Service Bldg., Milwaukee, Wis.

CARPENTER, Rolla C. (1909), Prof. of Expl. Engrg., Sibley College, Cornell Univ., and 125 Eddy St., Ithaca, N. Y.

COLBY, Albert Ladd (1909), Cons. Engr., and Iron and Steel Metallurgist, 447 Lehigh St., S. Bethlehem, Pa.

DALLETT, W. P. (1909), Hydr. Engr. and Contr., 49 N. 7th St., Philadelphia, and Media, Pa.

FAIR, William J. (Affiliate, 1909), Engr., Murray Hill Hotel, New York, N. Y.

HERSCHEL, Winslow H. (Affiliate, 1909), Cons. Engr., Providence Engrg. Wks., 110 Governor St., Providence, R. I.

HULL, A. B. (Affiliate, 1909), Salesman, Fairbanks, Morse Co., 30 Church St., New York, N. Y.

HUNT, Charles B. (1909), Supt. of Constr., Buckeye Eng. Co., and 636 McKinley Ave., Salem, O.

HUSSEY, Wm. Edgerly (1909), Mgr. New York Office, Providence Engrg. Wks., 50 Church St., and 67 W. 92d St., New York, N. Y.

LEHN, Henry Coe (Affiliate, 1908), Draftsman, Williamsville, N. Y.

LYMAN, James (1909), Engr., Western Dist., Genl. Elec. Co., 1047 Monadnock Bldg., Chicago, and 1308 Maple Ave., Evanston, Ill.

MITCHELL, Charles J. (1909), Charge of Design, Fairbanks, Morse Mfg. Co., and *for mail*, 836 College Ave., Beloit, Wis.

O'BRIEN, Thomas (Affiliate, 1909), Mgr., Fulton Motor Car Co., 370 Gerard Ave., and *for mail*, 3217 Decatur Ave., New York, N. Y.

ODE, Randolph Theodore (1909), Secy., Providence Engrg. Wks., Providence, and 557 Fruit Hill Ave., North Providence, R. I.

RATHBUN, Edward (1909), 2d V. P., Rathbun-Jones Engrg. Co., Spencer St., Toledo, O.

ROBERTS, Edmund W. (1909), Mech. Engr., V. P. and Genl. Mgr., The Roberts Motor Co., Sandusky, O.

SCHEFFLER, Frederick A. (1909), The Babcock & Wilcox Co., 85 Liberty St., New York, N. Y., and *for mail*, 293 Ridgewood Ave., Glen Ridge, N. J.

SYKES, George (Affiliate, 1908), Engr. and Builder, 1123 Broadway, New York, N. Y.

THOMAS, Richard H. (Affiliate, 1909), 107 Liberty St., New York, N. Y.

WEEKS, Chas. H. (1909), V. P., Buckeye Eng. Co., Lock Box 175, and 301 McKinley Ave., Salem, O.

COMING MEETINGS

April 10 to May 10

AERONAUTIC SOCIETY
April 14, etc., evenings, weekly meetings, Automobile Club of America, W. 54th St., New York. Secy., Wilbur R. Kimball.

AIR BRAKE ASSOCIATION
May 11–14, annual meeting, Richmond, Va. Secy., F. M. Nellis, 53 State St., Boston, Mass.

AMERICAN ASSOCIATION ELECTRIC MOTOR MANUFACTURERS
May 17–20, annual meeting, Hot Springs, Va.

AMERICAN ELECTROCHEMICAL SOCIETY
May 5–8, annual meeting, Niagara Falls, Ont. Secy., Dr. J. W. Richards, Lehigh University, So. Bethlehem, Pa.

AMERICAN FOUNDRYMEN'S ASSOCIATION
May 18–20, Hotel Sinton, Cincinnati, O. Secy., Richard Moldenke, Watchung, N. J.

AMERICAN GAS POWER SOCIETY
April 27, quarterly meeting, Minneapolis, Minn. Secy., R. P. Gillette.

AMERICAN GEOGRAPHICAL SOCIETY
April 20, 29 W. 39th St., New York, 8 p.m. Secy., Geo. H. Hurlbut.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS.
April 9, monthly meeting, 33 W. 39th St., New York, 8 p.m. Subject: Engineering Education. Secy., R. W. Pope.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS
April 16, monthly meeting, Toronto section. Secy. *pro tem.*, W. H. Eisenheis, 1207 Traders' Bank Bldg.

AMERICAN MATHEMATICAL SOCIETY
April 24, Columbia University, New York. Chicago section, April 9, 10, regular meeting, University of Chicago. Secy., H. E. Slaught, 58th St. and Ellis Ave., Chicago.

AMERICAN PORTLAND CEMENT MANUFACTURERS
April 12–14, quarterly meeting, Bellevue-Stratford Hotel, Philadelphia, Pa. Secy., Percy H. Wilson, Land Title Bldg.

AMERICAN RAILWAY ASSOCIATION
May 19, annual meeting, New York. Secy., W. F. Allen, 24 Park Pl.

AMERICAN SOCIETY OF CIVIL ENGINEERS
April 21, May 5, semi-monthly meetings, 220 W. 57th St., New York. Secy., C. W. Hunt.

AMERICAN SOCIETY OF HUNGARIAN ENGINEERS AND ARCHITECTS
May 1, 29 W. 39th St., New York, 8.30 p.m. Pres., H. Pickler.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS
April 13, monthly meeting, 29 W. 39th St., New York; May 4–7, Spring Meeting, Washington, D. C. Secy., C. W. Rice, 29 W. 39th St.

ARCHITECTURAL INSTITUTE OF CANADA
April 6, special General Meeting, Toronto. Secy., Alcide Chaussé, Montreal.

BLUE ROOM ENGINEERING SOCIETY
May 6, 29 W. 39th St., New York, 8 p.m. Secy., W. D. Sprague.

BOSTON SOCIETY OF CIVIL ENGINEERS
April 21, monthly meeting, Tremont Temple. Secy., S. E. Tinkham, 60 City Hall.

BROOKLYN ENGINEERS' CLUB
April 1, 197 Montague St. Paper: The Pennsylvania Railroad Tunnel, J. H. O'Brien and Schuyler Hazard. Secy., Joseph Strachan.

CANADIAN FREIGHT ASSOCIATION
April 9, annual meeting. Secy., T. Marshall, Toronto.

CANADIAN RAILWAY CLUB
May 4, monthly meeting, Windsor Hotel, Montreal, 8 p.m. Secy., Jas. Powell, St. Lambert, Montreal.

CANADIAN SOCIETY OF CIVIL ENGINEERS, Manitoba Branch
May 6, monthly meeting, University of Manitoba. Secy., E. Brydone Jack.

CANADIAN SOCIETY OF CIVIL ENGINEERS, Quebec Branch
April 16, General Section Meeting; April 23, electrical section; April 30, business meeting; May 7, mechanical section, 413 Dorchester St., W., Montreal. Secy., Prof. C. H. McLeod.

CANADIAN SOCIETY OF CIVIL ENGINEERS, Toronto Branch
April 22, regular meeting, 96 King St., W. Secy., T. C. Irving, Jr.

CAR FOREMEN'S ASSOCIATION OF CHICAGO
April 12, May 10, monthly meetings. Secy., Aaron Kline, 326 N. 50th St.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
April 13, Cincinnati, O., 11 a.m. Secy., O. G. Fetter.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
April 14, Columbus, O.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
May 3, Indianapolis, Ind. Secy., G. B. Staats, care Penna. Lines.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
April 12, Kansas City, Mo. Secy., F. H. Ashley, Gumbel Bldg.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
April 20, Peoria, Ill.

CENTRAL RAILWAY AND ENGINEERING CLUB OF CANADA
April 20, monthly meeting, Rossin House, Toronto, Ont. Secy., C. L. Worth, Room 409, Union Sta.

CENTRAL RAILWAY CLUB
May 14, monthly meeting, Hotel Iroquois, Buffalo, N. Y., 8 p.m. Secy., H. D. Vought, 95 Liberty St., New York.

CLEVELAND ENGINEERING SOCIETY
April 13, monthly meeting, Caxton Bldg. Paper: Power Plant Equipment, F. W. Ballard. Secy., Joe C. Beardsley.

COLORADO SCIENTIFIC SOCIETY
May 1, monthly meeting, Denver. Secy., Dr. W. A. Johnston, 801 Symes Bldg.

EASTERN RAILROAD ASSOCIATION
May 13, annual meeting. Secy., John J. Harrower, 614 F St., N.W., Washington, D. C.

ENGINEERING ASSOCIATION OF THE SOUTH
April 20, monthly meeting, Nashville section, Carnegie Library Bldg. Section Secy., H. H. Trabue, Berry Blk.

ENGINEERING SOCIETY OF THE STATE UNIVERSITY OF IOWA
May 4, monthly meeting, Iowa City. Secy., Dean Wm. G. Raymond.

ENGINEERS' AND ARCHITECTS' CLUB
April 19, monthly meeting, 303 Norton Bldg., Louisville, Ky. Secy., Pierce Butler.

ENGINEERS' CLUB OF BALTIMORE
May 1, monthly meeting. Secy., R. K. Compton, City Hall.

ENGINEERS' CLUB OF CENTRAL PENNSYLVANIA
May 4, monthly meeting, Gilbert Bldg., Harrisburg. Secy., E. R. Dasher.

ENGINEERS' CLUB OF CINCINNATI
April 15, monthly meeting, 25 E. 8th St. Secy., E. A. Gast, P. O. Box 333.

ENGINEERS' CLUB OF PHILADELPHIA
April 17, May 1, semi-monthly meetings, 1317 Spruce St. Secy., H. G. Perring.

ENGINEERS' CLUB OF TORONTO
April 15, etc., weekly meetings, 96 King St., W. Secy., R. B. Woolsey.

ENGINEERS' SOCIETY OF MILWAUKEE
April 14, monthly meeting, 456 Broadway. Secy., W. Fay Martin.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA
April 20, regular meeting; May 4, sectional meeting. Secy., E. K. Hiles.

EXPLORERS' CLUB
May 7, 29 W. 39th St., New York, 8.30 p.m. Secy., H. C. Walsh.

GENERAL SUPERINTENDENTS' ASSOCIATION OF CHICAGO
April 14, Chicago. Secy., H. D. Judson, C. B. & Q. R. R.

ILLUMINATING ENGINEERING SOCIETY
April 8, May 13, monthly meetings, New York section, 29 W. 39th St., 8 p.m. Secy., P. S. Millar.

INTERNATIONAL MASTER BOILER-MAKERS' ASSOCIATION
April 27–30, convention, Hotel Sealbach, Louisville, Ky. Secy., H. D. Vought, 95 Liberty St., New York. Standardizing Blue Prints for Building Boilers; Boiler Explosions; Best Method of Applying Flues, Best Method for Caring for Flues While Engine is on the Road and at Terminals, and Best Tools for Same; Flexible Staybolts Compared with Rigid Bolts; Best Method of Applying and Testing Same; Steel vs. Iron Flues, What Advantage and What Success in Welding Them; Best Method of Applying Arch Brick; Standardizing of Shop Tools; Standardizing of Pipe Flanges for Boilers and Templets for Drilling Same; Which is the long way of the Sheet; Best Method of Staying the Front Portion of Crown Sheet on Radial Top Boiler to Prevent Cracking of Flue Sheet in Top Flange; Rules and Formulas; Senate Bill.

IOWA ELECTRICAL ASSOCIATION
April 21, 22, annual meeting, Cedar Rapids. Secy., W. N. Keiser, Des Moines.

IOWA RAILWAY CLUB
April 9, May 14, monthly meetings, Des Moines. Secy., W. B. Harrison, Union Sta.

LOUISIANA ENGINEERING SOCIETY
April 12, May 10, monthly meetings, 323 Hibernia Bldg., New Orleans. Secy., L. C. Datz.

MASSACHUSETTS STREET RAILWAY ASSOCIATION
April 14, monthly meeting, Boston. Secy., Chas. S. Clark, 70 Kilby St.

MISSOURI ELECTRIC LIGHT, GAS AND RAILWAY ASSOCIATION
April 15–17, annual convention, Springfield. Secy., C. L. Clary, Sikeston.

MODERN SCIENCE CLUB
April 6, regular meeting, 125 S. Elliott Pl., Brooklyn, N. Y. Paper: Electric Motor Design, C. A. Lundell. April 13, annual election. Secy., Jas. A. Donnelly.

MUNICIPAL ENGINEERS OF THE CITY OF NEW YORK
April 28, 29 W. 39th St., 8.15 p.m. Secy., C. D. Pollock.

MUSURGIA SOCIETY
April 15, 29 W. 39th St., New York, 8 p.m. Secy., F. M. Frobisher.

NATIONAL ASSOCIATION OF AUTOMOBILE MANUFACTURERS
May 5, monthly meeting, New York. Secy., C. C. Hildebrand, 7 E. 42d St.

NATIONAL ASSOCIATION OF COTTON MANUFACTURERS
April 28, 29, annual meeting, Mechanics Bldg., Boston, Mass. Secy., C. J. H. Woodbury, P. O. Box 3672.

NATIONAL ASSOCIATION OF MANUFACTURERS
May 11, annual meeting, New York. Secy., Geo. S. Boudinot, 170 Broadway.

NATIONAL FIRE PROTECTION ASSOCIATION
May 25–27, annual meeting, New York. Secy., W. H. Merrill, 382 Ohio St., Chicago, Ill.

NATIONAL RAILWAY WATER SUPPLY ASSOCIATION
April 11, Minneapolis, Minn. Secy., F. W. Hayden, Glencoe.

NATURAL GAS ASSOCIATION OF AMERICA
May 18–20, Columbus, O. Secy., J. F. Owens, Wagoner, Okla.

NEW ENGLAND RAILROAD CLUB
April 13, monthly meeting, Copley Square Hotel, Boston, Mass., 8 p. m. Paper: Smoke Prevention in Relation to Combustion, George F. Baker. Secy., Geo. H. Frazier, 10 Oliver St.

NEW ENGLAND STREET RAILWAY CLUB
April 22, monthly meeting, American House, Boston, Mass. Secy., John J. Lane, 12 Pearl St.

NEW YORK RAILROAD CLUB
April 16, monthly meeting, 29 W. 39th St., 8.15 p.m. Secy., H. D. Vought, 95 Liberty St.

NEW YORK SOCIETY OF ACCOUNTANTS AND BOOKKEEPERS
April 13, etc., weekly meetings, 29 W. 39th St., 8 p.m. Secy., T. L. Woolhouse.

NEW YORK TELEPHONE SOCIETY
April 20, monthly meeting, 29 W. 39th St., 8 p.m. Secy., T. H. Laurence.

NORTHERN RAILWAY CLUB
April 24, monthly meeting, Commercial Club Rooms, Duluth, Minn. Secy., C. L. Kennedy.

NORTHWEST RAILWAY CLUB
April 13, May 11, monthly meetings, Minneapolis, Minn. Secy., T. W. Flannagan, care Soo Line.

NOVA SCOTIA SOCIETY OF ENGINEERS
April 8, May 13, monthly meetings, Halifax. Secy., S. Fenn.

OPTOMETRICAL SOCIETY OF THE CITY OF NEW YORK
April 14, 29 W. 39th St., 8 p.m. Secy., J. H. Drakeford.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS
April 27, monthly meeting, Technical High School Hall, 8 p.m. June 22, annual meeting. Secy., T. M. Phetteplace.

PURDUE MECHANICAL ENGINEERING SOCIETY
April 14, etc., fortnightly meetings, Purdue University, Lafayette, Ind., 6.30 p.m. Secy., L. B. Miller.

RAILWAY CLUB OF KANSAS CITY
April 16, Kansas City, Mo.

RAILWAY CLUB OF PITTSBURGH
April 23, monthly meeting, Monongahela House, 8 p.m. Secy., J. D. Conway, Genl. Office, P. & L. E. R. R.

RAILWAY SIGNAL ASSOCIATION
May 11, Chicago, Ill. Secy., C. C. Rosenberg, 712 North Linden St., Bethlehem, Pa.

RENSSELAER SOCIETY OF ENGINEERS
April 23, etc., fortnightly meetings, 257 Broadway, Troy, N. Y. Secy., R. S. Furber.

RICHMOND RAILROAD CLUB
April 12, Richmond, Va. Secy., F. O. Robinson, care C. & O. R. R.

ROCHESTER ENGINEERING SOCIETY
April 9, May 14, monthly meetings. Secy., John F. Skinner, 54 City Hall.

ST. LOUIS RAILWAY CLUB
April 9, monthly meeting, Southern Hotel, 8 p.m. Secy., B. W. Frauenthal.

SCRANTON ENGINEERS' CLUB
April 15, monthly meeting, Board of Trade Bldg. Secy., A. B. Dunning.

SHORT LINE RAILROAD ASSOCIATION
May 3, New York. Secy., Cromwell G. Macy, Jr., Nantucket Central R. R., 257 Broadway.

SOUTHERN AND SOUTHWESTERN RAILWAY CLUB
April 15, monthly meeting, Atlanta, Ga. Secy., A. J. Merrill, 218 Prudential Bldg., Atlanta.

SOUTHERN ASSOCIATION OF CAR SERVICE OFFICERS
April 15, Atlanta, Ga.

SOUTHWESTERN ELECTRICAL AND GAS ASSOCIATION
May, annual convention, Dallas, Texas.

TECHNICAL SOCIETY OF BROOKLYN
April 16, semi-monthly meeting, Arion Hall, Arion Pl., 8.30 p.m. Pres., M. C. Budell, 20 Nassau St., New York.

TECHNOLOGY CLUB OF SYRACUSE
April 14, May 12, monthly meetings, 502 Bastable Blk., 8 p.m. Secy., Robert L. Allen.

WESTERN RAILWAY CLUB
April 20, monthly meeting, Auditorium Hotel, Chicago, Ill., 8 p.m. Secy., Joseph W. Taylor, 390 Old Colony Bldg.

WESTERN SOCIETY OF ENGINEERS

April 21. Paper: Protective Coatings for Structural Materials, R. S. Perry. April 14, Electrical section, Chicago, Ill. Secy., T. H. Warder, 1737 Monadnock Blk., Chicago.

WISCONSIN GAS ASSOCIATION

May 12, 13, annual convention, Milwaukee. Secy., Henry H. Hyde, Racine.

MEETINGS TO BE HELD IN ENGINEERING SOCIETIES BUILDING

Date April	Society	Secretary	Time
13	N. Y. Society Accountants and Bkprs	T. L. Woolhouse	8:00
13	American Society Mechanical Engineers	C. W. Rice	8:00
14	Optometrical Society of City of N. Y	J. H. Drakeford	8:00
15	Musurgia Society	F. M. Frobisher	8:00
16	New York Railroad Club	H. D. Vought	8:15
20	American Geographical Society	Geo. H. Hurlbut	8:00
20	N. Y. Society Accountants and Bkprs	T. L. Woolhouse	8:00
20	New York Telephone Society	T. H. Laurence	8:00
27	N. Y. Society of Accountants and Bkprs	T. L. Woolhouse	8:00
28	Municipal Engineers of New York	C. D. Pollock	8.15
May			
1	Amer. Soc. Hungarian Engrs & Archts	H. Pickler (Pres.)	8:30
4	N. Y. Society Accountants and Bkprs	T. L. Woolhouse	8:00
6	Blue Room Engineering Society	W. D. Sprague	8:00
7	Explorers Club	H. C. Walsh	8:30

NEW BOOKS

Water Power Engineering. The Theory, Investigation and Development of Water Powers. By Daniel W. Mead, Consulting Engineer; Professor of Hydraulic and Sanitary Engineering, University of Wisconsin. *McGraw Publishing Co., New York*, 1908. 8vo, cloth, viii + 787 pp. Price, $6.00 net.

In the development of a water power project, the engineer is frequently called upon to do more than design and construct the power plant. He has to consider the adequacy of the supply, the power available, the cost, and many other preliminary problems, and the author has therefore discussed at length the fundamental principles which must form the basis of a successful water power development. Much the smaller part of the book is devoted to the theory of hydraulics and of design and much the larger part to the broader considerations of water power development, turbine analysis and selection, and other controlling features upon which but little positive information has usually been available. In this connection those who attended the Detroit meeting of the Society will remember the excellent discussion upon surge tanks by Mr. L. F. Harsa, which he then stated was based upon material prepared by him under the direction of Professor Mead for use in this book which was then in process. This matter is included in the chapter on Governing, and it may be taken as a fair indication of the thoroughness with which the author has worked up the various chapters. At the end of each chapter is a list of literature upon the subjects covered, affording the student and engineer an opportunity to investigate further any subject in which he is interested.

Contents, by chapter headings: Introduction; Power; Hydraulics; Water Power; Rainfall; The Disposal of the Rainfall; Run-off; Stream Flow; The Measurement of Stream Flow; Water Wheels; Turbine Details and Appurtenances; Hydraulics of the Turbine; Turbine Testing; The Selection of the Turbine; The Load Curve and Load Factors, and Their Influence on the Design of the Power Plant; The Speed Regulation of Turbine Water Wheels; The Water Wheel Governor; Arrangement of the Reaction Wheel; The Selection of Machinery and Design of Plant; Examples of Water Power Plants; The Relation of Dam and Power Station; Principles of Construction of Dams; Appendages to Dams; Pondage and Storage; Cost, Value and Sale of Power; The Investigation of Water Power Projects; Appendices.

The Temperature-Entropy Diagram. By Charles W. Berry, Assistant Professor of Mechanical Engineering, Massachusetts Institute of Technology. *John Wiley & Sons, New York*. 1908. Second edition revised and enlarged. Cloth, 12mo, xviii + 299 pp. 109 figures. Price, $2.00 net.

In order to use this little book, the reader must have previously been a student of thermo-dynamics, but to such as have a good acquaintance with the subject it will prove a useful handbook on practically the whole range of subjects covered by the science. The treatment is both by temperature-entropy diagram and by formula, the explanation of the diagrams and formulas following along together, thus making plain to the reader the heat changes that actually occur as well as

giving him the means for calculating the changes numerically. To show the method of treatment and the degree of completeness, reference may be made to a single chapter—that of The Flow of Fluids. This takes up by diagram and by formula, on the principle of the transfer of energy, the flow of gases, of saturated and superheated steam, design of nozzles, loss of availability through throttling, the throttling calorimeter, the principles by which the specific heat of superheated steam is determined, adiabatic expansion with and without friction loss, etc. In the present edition, a chapter is given explaining Mollier's total energy-entropy diagram.

Contents, by chapter headings: General Discussion; The Temperature-entropy Diagram for Perfect Gases; The Temperature-entropy Diagram for Saturated Steam; The Temperature-entropy Diagram for Superheated Vapors; The Temperature-entropy Diagram for the Flow of Fluids; Mollier's Total Energy-entropy Diagram; Thermodynamics of Mixtures of Gases and Vapors, and of Vapors; The Temperature-entropy Diagram Applied to Hot-Air Engines; The Temperature-entropy Diagram Applied to Gas-engine Cycles; The Gas-engine Indicator Card; The Temperature-entropy Diagram Applied to the Non-conducting Steam-engine; The Multiple-fluid or Waste heat Engine; The Temperature-entropy Diagram of the Actual Steam-engine Cycle; Steam-engine Cylinder Efficiency; Liquefaction of Vapors and Gases; Application of the Temperature-entropy Diagram to Air-compressors and Air-motors; Discussion of Refrigerating Processes and the Warming Engine; Table of Properties of Saturated Steam from 400° F., to the Critical Temperature.

THE MECHANICAL ENGINEERING OF STEAM POWER PLANTS. By Frederick Remsen Hutton, E.M., Sc.D. *John Wiley & Sons, New York.* 1908. Third edition, rewritten, Cloth, 8vo, xli + 825 pp., 697 illustrations. Price $5.

The first edition of this work was issued in 1897 and filled a new place in literature upon the power plant. The treatise was prepared for the business engineer, the contractor, and the man who buys or installs machinery and is interested in power-plant economy as affected by the relations of one unit to another, rather than by the minute details of design. It contained a comprehensive description of all the apparatus of the power plant and it had as a characteristic feature a statement of the advantages or disadvantages of the different types of apparatus and of different methods of operation, with a view to placing before the reader the fundamental considerations in power plant equipment. In the new edition the same plan is followed, but amplified to cover more important power house problems and more attention is given to the principles of mechanics related to the subject. This edition has been entirely rewritten, with much new matter, and its object, as outlined above, is further indicated by the statement in the preface to the effect that while there are few engineers called upon to design and construct any detail of the power plant, all are sure to be called upon to buy some or all of these elements or to design power plants as a whole in which such elements are to function. Hence the importance of training in the process of selection and of critically weighing arguments for and against apparatus.

Contents, by chapter headings: *Part I.* Introductory; The Function of the Power Plant; Sources of Motor Energy for the Power Plant; Internal and External Combustion; Measurements of the Work Unit of Output; Indicated Horsepower; Brake Horsepower; Elements and Analysis of the Steam Power Plant; The Quantitative Basis of the Steam Power Plant. *Part II.* The Boiler: Forms, Material and Manufacture; Boiler; Riveting, Staving and Structural Details: Fire-Tube Boilers Externally Fired; Fire-Tube Boilers Internally Fired; Water Tube Boilers; Coil and Pipe Boilers; Flash and Semi-Flash Boilers; Boiler Furnaces, Chimneys and Settings;

Firing Boilers with Gas or Liquid Hydro-carbon or with Pulverized Fuel; Boiler Accessory Apparatus; Care and Management of Boilers; Boiler Inspection and Testing; Boiler Explosions. *Part III.* Boiler Plant Auxiliaries. *Part IV.* The Piping of Pressure to the Engine and its Accessories. *Part V.* The Engine; Expansive Working of Steam; The Compound and Multiple Expansion Engine; The Rotary Steam Engine; The Steam Turbine; Engine Foundation and Bed-Plate; Engine Cylinder, Piston and Piston-Rod; Cross-Head, Guides and Connecting-Rod; Crank-Shaft, Eccentric, Fly-Wheel; Valves and Valve Gearing; Valve Gearing Design; Special Forms; Valve Gearing; Balanced Valves; Cam and Trip Valve-Gear; Reversing Valve Gears; Link Motions; Valve Gears for Variable Cut-off; Governing and Governors for Steam Engines. *Part VI.* Engine Auxiliaries: The Condenser and Attachments; Engine Auxiliaries, Lubricators and Lubrication. *Part VII.* Care and Management; Accidents; Testing the Power Plant for Economy and Efficiency. *Part VIII.* General Remarks upon the Power Plant. *Part IX.* Appendices; Historical Summary; Steam Tables; Table Hyperbolic Logarithms; Historic Illustrations.

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G. T. Voorhees
A. P. Trautwein
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On Revision and Extension of the Code for Testing Gas Power Machinery

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On Hudson-Fulton Celebration

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